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S·A·E JOURNAL

DECEMBER 1932

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A RIDING COMBINATION HITHERTO UNKNOWN IN MOTORING

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This new contribution to riding and driving ease and safety will be acclaimed by every automotive engineer, every automotive executive, every automobile dealer, and every speed-loving motorist in America.

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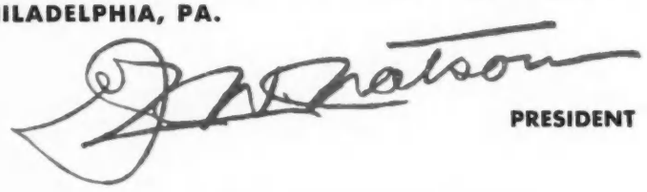
PHILADELPHIA, PA.



PRESIDENT

GYRO

FIRM AGAINST SIDE SWAY

S·A·E· JOURNAL

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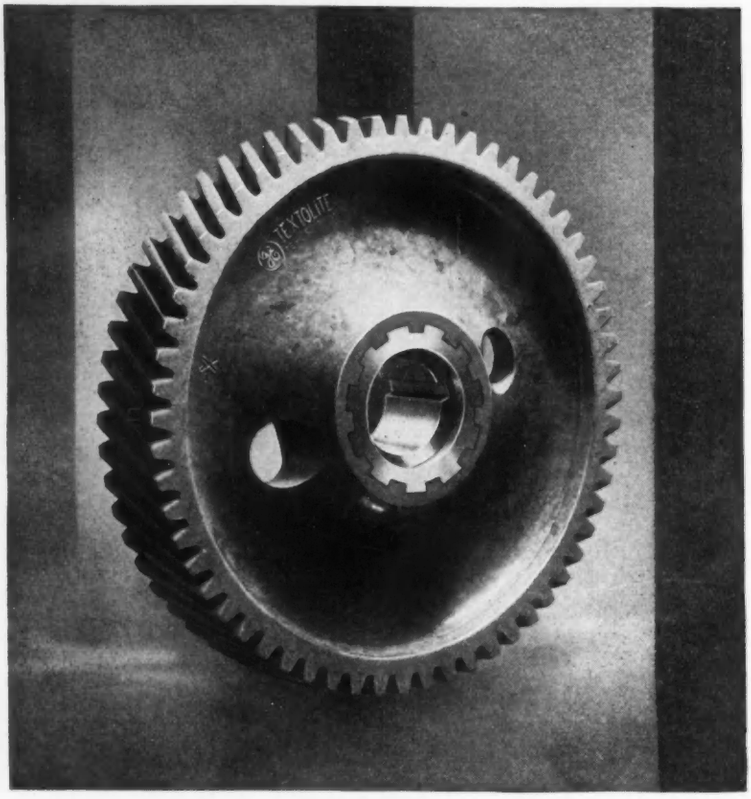
NOTE: Page numbers above that are preceded by (T) refer to the TRANSACTIONS section (between pp. 16 and 17 of the advertising and news section) containing papers and discussions that will be embodied in the volume of S.A.E. TRANSACTIONS for 1932, to be issued early in 1933.

The purpose of meetings of the Society is largely to provide a forum for the presentation of straightforward and frank discussion. Discussion of this kind is encouraged. However, owing to the nature of the Society as an organization, it cannot be responsible for statements or opinions advanced in papers or in discussions at its meetings. The Constitution of the Society has long contained a provision to this effect.



NO Cumulative Wear

THE purpose of any timing mechanism is to maintain a definite phase relation between crankshaft and camshaft. The more the wear, the more the camshaft is out of time.

With timing gears, the displacement is confined to the wear on the tooth profile. There is no cumulative wear.

To-day's nonmetallic gear material is so made that average wear is less than .001 in. in 20,000 miles of service—an amount so small that camshaft lag is infinitesimal.

TEXTOLITE TIMING GEARS

Textolite timing gears are good for 50,000 miles—and in numerous cases have given upwards of 100,000 miles of quiet service.

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GENERAL ELECTRIC

Advance News of January Events

BANCROFT GHERARDI, engineering vice-president of the American Telephone & Telegraph Co., and Roy Faulkner, president of the Pierce-Arrow Sales Corp., have consented to be the speakers of the evening at the Society's Annual Dinner Jan. 12, 1933, at the Hotel Pennsylvania, New York City, and all who have had the good fortune to hear either or both of these distinguished men will immediately realize that the success of the Dinner is already assured. The subjects of Mr. Gherardi's and Mr. Faulkner's addresses will be announced later.

A singular piece of good luck enables the Dinner Committee to announce that Reinald Werrenrath, famous baritone, has graciously consented to sing at the Dinner.

An amusing feature of the evening will be a more or less scientific discourse on automotive matters, offered by Rube Goldberg, the so-called inventor of the balloon tire.

Railroad Fares Reduced

Members who will travel to New York City to attend the Dinner and to Detroit for the Annual Meeting will be glad to know that the railroad companies have agreed to grant a reduction in fares to S.A.E. members for these events. Details will be printed in the next Bulletin and in the January issue of THE JOURNAL.

The Annual Meeting

The forthcoming Bulletin, as well as the January JOURNAL, will contain a complete program of the Annual Meeting which will be held at the Book-Cadillac Hotel, Detroit, Jan. 23 to 26.

Topics to be discussed in the papers have been chosen with great care by experienced committee members who recognize the value of timeliness in such matters. Moreover, the engineers who, at the invitation of the respective committees, have agreed to prepare and deliver these papers are men whose opinions are based on broad experience and intensive study of their subjects.

The list of "high spots" appearing on this page is but a foretaste of the complete schedule of papers to be announced later.

High Spots of the Meeting

Constructive Criticism of Present Passenger-Car Design—Herbert Chase, New York City.

Rubber-Tired Rail Coaches—Col. E. J. W. Ragsdale, Edward G. Budd Mfg. Co.

A Dinner, on Thursday evening, Jan. 26, sponsored by the Detroit Section.

FOR STUDENTS OF ALL AGES

Sixty Octaves of Radiant Energy—Dr. H. C. Rentschler, Westinghouse Lamp Co.

The Automobile's Place in American Transportation—B. E. Hutchinson, Chrysler Corp.

ENGINE PROBLEMS, VARIED AND VITAL

Aluminum Cylinder-Heads—F. F. Kishline, Graham-Paige Motors Corp.

Testing with the Farnboro Electric Indicator—E. T. Vincent, Continental Aircraft Engine Co.

Fuels for High-Speed Oil Engines—Max Hofmann, Waukesha Motor Co.

Indirect Air-Cooling of Aviation Engines—J. H. Geisse, Madison, Wis.

The Powerplant in Airplane Flight Testing—Two or three papers by members of the United Aircraft & Transport and the Pratt & Whitney Aircraft groups.

WHAT ABOUT BODIES AND CHASSIS?

A New Method of Measuring the Air Resistance of Motor-Vehicle Bodies—W. E. Lay, University of Michigan.

What Price Standardization?—W. T. Fishleigh, Detroit.

Automatic Transmissions—W. C. Keys, Detroit.

The Future of Low-Pressure Balloon Tires—A. F. Denham, The Chilton Co.

The New A-B-C in Clutch Design—E. E. Wemp, Long Mfg. Co.

Final Report on Riding Qualities—Dr. F. A. Moss, George Washington University.

Inertia-Control Shock-Absorbers—Carl Kindl, Delco Products Corp.

PRODUCTION MEN AT LUNCH AND LATER

Some Principles of Flexible, Accurate, Low-Cost Tooling—J. E. Padgett, Spicer Mfg. Corp.

Production, a Human Equation—K. T. Keller, Chrysler Corp.

Automotive Aspects of Work of the Committee on Rehabilitation of Industry—Col. J. L. Walsh, Chairman, Michigan District, National Rehabilitation Committee.

COOPERATIVE FUEL-RESEARCH DATA

The Vapor-Handling Capacity of Automobile Fuel-Systems — O. C. Bridgeman, H. S. White and F. B. Gary, Bureau of Standards.

The Load-Carrying Capacity of Extreme-Pressure Lubricants—S. A. McKee, Bureau of Standards.

Friction Data in the Region of Thin-Film Lubrication—O. C. Bridgeman, Bureau of Standards.

Correlation of C.F.R. Laboratory Knock Ratings with Behavior of Motor Fuels in Service—C. B. Veal, Society of Automotive Engineers; H. W. Best, Yale University; J. M. Campbell, General Motors Corp. Research Laboratories, and W. M. Holaday, Standard Oil Co. (Indiana).

Meetings Calendar

General Meetings

Annual Dinner—Jan. 12, 1933

Hotel Pennsylvania, New York City

Annual Meeting—Jan. 23 to 26, 1933

Book-Cadillac Hotel, Detroit

Section Meetings

Baltimore—Dec. 15

Hotel Emerson; Dinner 6:30 P.M.
Lubricants

Canadian—Dec. 14

Royal York Hotel, Toronto; Dinner 6:30 P.M.
Spark Plugs—F. E. Howard, A. C. Spark Plug Co.

Chicago—Dec. 6

Hotel Sherman; 8:00 P.M.
Recent Developments in Petroleum Industry—Robert E. Wilson, Standard Oil Co. (Indiana)

Cleveland—Dec. 12

Cleveland Club; Dinner 6:30 P.M.
Low-Pressure Tire Problems, from the Viewpoint of the Chassis Engineer—Tore Franzen, Chrysler Corp.; from the Viewpoint of the Tire Manufacturer—B. J. Lemon, United States Rubber Co.

Dayton—Dec. 15

Engineers' Club; Dinner 6:30 P.M.
Brakes and Automatic Clutches—By a representative of the Bendix Brake Co.

Detroit—Dec. 5

Book-Cadillac Hotel; Dinner 6:30 P.M.

Indiana—Dec. 8

Indianapolis Athletic Club; Dinner 6:30 P.M.
Causes of Wheel Wabble, Shimmy and Tramp—Dr. O. E. Kurt, United States Rubber Co.

Kansas City—Dec. 10

LaSalle Hotel; Dinner 6:30 P.M.
Modern Motor Fuels—Gustav Egloff, Universal Oil Products Co.
Demonstration of antiknock fuels with portable testing machine

Metropolitan—Dec. 8

Hotel New Yorker, New York City; Dinner 6:15 P.M.
Engines—H. M. Crane, General Motors Corp.

Milwaukee—Dec. 9

Le Roi Co. Plant
Service and Laboratory Fuel-Testing—William M. Holaday, Standard Oil Co. (Indiana)

New England—Dec. 14

Walker Memorial, Massachusetts Institute of Technology, Cambridge, Mass.; Dinner 6:30 P.M.
Downdraft Carburetion—Thomas Milton

Northern California—Dec. 13

Athens Athletic Club, Oakland; Dinner 6:30 P.M.
Diesel Engines and Fuels

Northwest—Dec. 2

Bergonian Hotel, Seattle; Dinner 6:30 P.M.
Gears—Thomas J. Bannan, Western Gear Works
Grease—George A. Zamboni, Sinclair Refining Co.

Oregon—Dec. 2

Multnomah Hotel, Portland; Dinner 6:30 P.M.

Philadelphia—Dec. 14

715 North Broad Street; Dinner 6:30 P.M.
Passenger-Car Transportation Meeting
Speaker: Charles O. Guernsey, J. G. Brill Co.

Pittsburgh—Dec. 6

Fort Pitt Hotel; Dinner 6:00 P.M.; Entertainment
Some Phases of Motor-Vehicle Legislation—A. J. Scaife, The White Motor Co.
Motor-Vehicle Tax Situation as It Applies Particularly to Pennsylvania—Virgil W. Thomas, John Beckert & Sons

Southern California—Dec. 2

Richfield Building, Los Angeles; Dinner 6:30 P.M.
Fuel for High-Compression Engines—David E. Day, Richfield Oil Co.

Syracuse—Dec. 5

Technology Club of Syracuse
Motor-Equipment Development, with Particular Reference to the Motorcoach—A. J. Scaife, The White Motor Co.

Washington—Dec. 21

Racquet Club; Dinner 6:30 P.M.
Fuels

A Challenge to Body-Design Progress

A. J. Neerken, S.A.E. Vice-President, Representing Passenger - Car - Body Engineering

BEAUTY, style, comfort are beginning to be recognized more and more as factors of prime importance in the successful design and sale of our product.

If we compare our car-body designs of today, both as to style and construction, with the designs of 10 or even 5 years ago, we are amazed at the progress that has been made. But I believe that we shall see far greater progress in the next decade.

Ever-increasing competition and the constantly more exacting demands of the buyer are challenging us to make such progress. With faith in the future of our profession, with faith in ourselves, with open and alert minds, with conscientious effort, we can meet this challenge.

We firmly believe that the opportunities for real achievement in our own particular field never were greater than they are today.

Cordially yours,

A. J. Neerken

Body Engineer,
Hupp Motor Car Corp.

Philadelphia Displaces Indiana

Membership Lead Now Held by Pennsylvanians

THE JOURNAL hesitates to become one of those prognosticators of the "I told you so" variety. Nevertheless, didn't the last issue predict an uprising of the Philadelphia Section as a result of being squeezed back into second place last month in the Section race for leadership in the Get-Your-Man contest?

The rebound from this situation has again carried the Philadelphia Section into the lead; not far, to be sure, but still in the lead. This has placed Indiana in second position, and the Canadian Section has stepped over Baltimore, making their standings respectively third and four places. The Metropolitan Section has leaped into sixth place, with Detroit following two behind, eighth in the list.

SECTION STANDING

(Expressed in Percentage of Quota)

1	Philadelphia	38.8
2	Indiana	35.7
3	Canadian	31.0
4	Baltimore	29.9
5	Southern California	26.0
6	Metropolitan	19.1
7	Chicago	18.7
8	Detroit	18.1
9	Kansas City	16.6
9	Northwest	16.6
10	Cleveland	12.3
11	Syracuse	12.1
12	Washington	12.0
13	Pittsburgh	11.6
14	Buffalo	10.5
15	St. Louis	9.7
16	Dayton	8.1
17	New England	7.8
18	Milwaukee	7.1
19	Northern California	6.0
20	Wichita	5.9
21	Oregon	0.0

In the individual standing, no very marked changes have occurred. John F. Hardecker, of the Philadelphia Section, still leads, with two members instead of four tied for second place.

Earlier in the contest it was anticipated that determination of the winners of the awards could be made by Dec. 31 and the results announced at the Annual Meeting in January in Detroit. However, owing to the careful consideration being given to each applicant and the work necessary in checking references and qualifications, it has been found impossible to present all of the applicants in the contest to the Council for election in time for the January announcement.

More time than was originally anticipated will be required, for a considerable number of applicants still remain to be presented for election. Therefore it is impossible now to determine an exact date on which announcement of the winners can be made. This will be easily understood when it is recalled that after election the newly elected member has 90 days in which to pay his initiation fee and dues and consummate his membership in the Society. A definite date will be announced as soon as the last application sent in during the Get-Your-Man contest has been acted upon by the Council.

The standing of the Sections and of the individuals is given herewith as of Nov. 21, 1932. What may happen next in the Section contest is anybody's guess.

Since the initial list of sponsors of new members was published, a number of new members have been received into the Society as a result of the membership campaign. Credit is extended to the following additional members who have sponsored one or more new members since the original list was published:

J. H. Albrecht
Roy F. Anderson
J. H. Ballard
C. H. Baxley
C. S. Crawford
William Engel
Randolph F. Hall
P. J. Kent
Arthur Nutt
W. H. Radford
C. O. Richards
Clifford Sklarek
E. S. Van Dusen
Loris R. Anderson
Percy J. Arrow
William L. Batt
Clarence S. Bruce
W. W. Dunnell, Jr.
C. E. Frudden
Raymond D. Kelly
Nelson G. Kling
Matt Payne
R. N. Reinhard
H. E. Simi
Max L. Tost

INDIVIDUAL STANDING

(For Individual Awards)

First Place

John F. Hardecker

Tied for Second Place

H. M. Jacklin — **L. M. Porter**

Tied for Third Place

R. N. DuBois — **F. K. Glynn**
A. Gelpke — **C. H. Jacobsen**
L. R. Joslin

Tied for Fourth Place

George B. Allen — **G. O. Pooley**
R. N. Janeway — **C. C. Stewart**
C. C. Mathis — **R. R. Teetor**
L. V. Newton — **O. M. Thornton**

Tied for Fifth Place

A. F. Coleman — **B. J. Lemon**
R. H. Combs — **Reese Lloyd**
Malcom R. Cox — **Charles I. MacNeil**
J. B. Franks, Jr. — **Arthur Nutt**
J. G. Holmstrom — **Edward W. Stock**
W. C. Keys — **E. E. Tattersfield**
C. G. Kreiger — **Alex Taub**
George W. Winter

REMEMBER A MEMBER

THE Society in its relation to its members can, in general, sum up its entire activities and focus them on one point—service. Whether it be service to an individual, to a group or to an industry, it is a cooperative effort of individual members as personified by its Committes and, in a greater sense, by its Sections.

In its employment work, the Society has ever been active, but only at the present time has such emphasis been needed in this work, and cooperative effort is the controlling factor.

Section cooperation in all activities of the Society has never been lacking and, as always, the Sections have turned their interests and resources toward joint effort in the matter of solving the employment problem among its members. The *Chicago Tribune* recently recognized the work of our Chicago Section in an article which states:

> Formation of an Employment Committee has been announced by Leonard V. Newton, Chairman of the Chicago Section of the Society of Automotive Engineers.
>
> The Chairman of the Committee is Robert T. Hendrickson, president of the Hendrickson Motor Truck Co. and a director of the Chicago Automobile Trade Association. Other members are C. A. Peirce, vice-president in charge of engineering and production of the Diamond T Motor Car Co.; Leo Huff, transportation engineer of the Pure Oil Co., and Harold Nutt, chief engineer of the Borg & Beck Co.

Subsequently, Employment Committees have been organized in the

Draftsmen
Fleet Operators
Metallurgists
Tool Designers
Maintenance Engineers
Inspectors
Body Designers
Aircraft Engineers
Production Engineers
Stress Analysts
Plant Superintendents
Technical Editors
Truck Designers
Fuel and Lubricant Engineers
Dynamometer Operators
Chassis Designers
Personnel Men
Transportation Engineers
Chemical Engineers
Airplane-Engine Designers
Consulting Engineers
Diesel-Engine Designers
Highway-Equipment Engineers
Road-Test Engineers
Sales Directors
Rail-Car Designers
Propeller Designers
Efficiency Engineers
Machinery Designers
Engineering Professors

Buffalo, Metropolitan, Milwaukee, Northwest, Pittsburgh and Southern California Sections to date, and similar action undoubtedly will be taken by many of the other Sections in forthcoming meetings, after discussion of the best way to approach the subject locally.

The Detroit Section, by virtue of its permanent office and Assistant Secretary, handles the work without a Committee.

Each individual member of the Society, no matter where employed, is a member of a great committee pledged to assist unemployed members in their search for placement. The problem is not so much the locating of unemployed members for whom jobs may be found as it is the finding of positions available. To this end the Sections are seeking the help of their members in an endeavor to promote the use of the S. A. E. Employment Service by employers generally throughout all branches of the industry.

Some time soon in all organizations in which our members are employed a need will arise for trained men capable of stepping into the organization and contributing satisfactorily to the success of the enterprise, and, by proper vigilance and cooperation between our members and the local Section organizations, it is believed that the Society's Employment Service can extend its usefulness over a wider field and to a greater number of its unemployed members.

REMEMBER A MEMBER!

S. A. E. EMPLOYMENT SERVICE

Chronicle and Comment

New Model Journal To Appear

AT a time when new automobile models are in the limelight, members of the Society will usher in a new and improved model of the S.A.E. JOURNAL. As authorized by our Council, the January issue will appear in a new cover and with an improved inside treatment. The first issue for 1933 will introduce a series of editions that will profit by the continued attention of recognized artists and publication men.

January Journal To Describe 1933 Vehicles

THROUGH the cooperation of Austin M. Wolf, THE JOURNAL for January will present an authoritative review of the engineering features of the new models that are to appear at the shows. In the preparation of his article Mr. Wolf is enjoying the full cooperation of engineering departments and technical men throughout the industry. The result will be, not a set of specifications but a truly significant and informative account of the recent engineering accomplishments of our own members and their associates.

Wide recognition in this Country and abroad has been accorded to similar articles by Austin Wolf that have appeared in THE JOURNAL, and it is certain that the 1933 review will meet with even greater favor.

A Great Attraction for Jan. 12

"BIGGER, better but less expensive" might well be the slogan for the Annual Dinner in New York City on Jan. 12.

Bancroft Gherardi, Engineering Vice-President of American Telephone & Telegraph Co.

Roy Faulkner, President of the Pierce-Arrow Sales Corp.

Reinald Werrenrath, renowned baritone and star of Music in the Air.

Rube Goldberg, cartoonist extraordinaire and so-called balloon-tire inventor.

What could be better?

Attend the Annual Meeting!

DESIGNERS of the major meeting of the year are applying the finishing touches to a program for Detroit in January that should bring out an exceptionally large attendance. The partial schedule, printed elsewhere in this issue, includes headline speakers who will present papers to inform, inspire and provoke discussion. Final program in the January issue. Reserve Jan. 23 to 26 for this event!

Met. Produces New Booster

UNDER the effective chairmanship of F. H. Dutcher, the Society's Metropolitan Section has set about establishing new records of performance and presenting attractive innovations. Already the Section's regular meetings have been supplemented by excellent sessions of the Marine and Aeronautic Divisions. And now the officers have produced a *Booster* which is indeed a super-publication.

Dr. A. E. Becker, in charge of the Oct. 20 meeting, is receiving congratulations on Vol. 1, No. 1, of the new *Booster*, a 12-page pamphlet which contains substantially the complete paper on Research that was presented by Past-President Horning, plus several contributions from discussers

Best wishes to the new *Booster* and its editors!

Horning on Research Next Month

FACT-FINDING was well established as the true basis of research by Past-President Horning, who regaled members of the Metropolitan Section with one of the most refreshing discourses ever-presented before the Society.

A wealth of Horning philosophy appears in the paper which will be printed in the S. A. E. JOURNAL for January.

Remember a Member!

PRESIDENT SCAIFE has established contact with many executives and employers in a further effort to bring S. A. E. members fully into line for consideration when openings occur for engineering talent. Assurances have come from important sources that Society members will be given every possible attention.

Cooperating with general activities of our Employment Service, numerous Sections have organized to make the work more nearly unanimous and more completely effective.

Here is an opportunity for the vast majority of our members, who are employed, to lend a friendly hand to those who lack positions. REMEMBER A MEMBER!

Cast-Iron Crankshafts Predicted

NOT so long ago, mention of a cast-iron crankshaft was good for a hearty laugh; today, engineers are working and wondering; tomorrow, the well-known C. I. will certainly be turning up proudly in many unexpected places. Right now, in fact, the new alloy irons are being assigned to new duties in the automotive structure, and the experimental laboratories are finding other possibilities.

In the course of his disquisition on the elimination of "lazy pounds" and related problems of design from the materials standpoint, Councilor H. T. Woolson, Chrysler chief engineer, recently told a Detroit Section audience how the iron alloys, just one step away from the blast furnace, have already improved engineering and manufacturing practices at such important points as the cylinder block, piston-ring, brake-drum, exhaust manifold, valve-stem guide and the like. He reported present production of cast camshafts and felt safe in predicting the development of crankshafts by casting methods.

Mentioning the engineers' longing for materials "yet unborn," Mr. Woolson took his hat off to metallurgical engineers whose wonderful past accomplishments are only indicative of great future possibilities.

Transportation Engineering

Body-Ventilating Problems Discussed

Subcommittee Report[1] Brings Out Ways of Reducing Odors and Freeing Vehicles from Them

PREPARED discussion presented at the Transportation Meeting in Toronto last October on the Subcommittee report on the important subject of motorcoach and motor-truck body ventilation follows. The sources of dangerous or disagreeable gases and fumes, measures taken to prevent or reduce them and methods of ventilating bodies to rid them of odors and supply temperature-regulated fresh air to passengers and truck drivers are discussed.

Engine the Source of Odors

H. R. HOLDER[2]:—While Mr. Cumming's paper contains some excellent features, I feel that we are all missing the main point and going round in circles on this very important question. Plenty of good ventilators on the market are suitable for giving the necessary ventilation to any commercial vehicle, but the way to avoid odors complained of is to prevent the engine from causing them; in other words, to attack the trouble at its source, which we all agree is the inlet manifold.

Manifold design is virtually the same now as it was 15 or 20 years ago, and the manufacturers give the operators the same manifold whether they are operating city lines with an average schedule speed of around 10 m.p.h. or interurban runs with a schedule speed of 20 to 40 m.p.h. This is wrong. Almost all of the trouble occurs during deceleration and acceleration and is due to the high vacuum set up in the manifold during deceleration and idling periods, which causes the manifold to load up. We have measured the vacuum under these circumstances and find that it runs as high as 26 in. We have been experimenting with a device for breaking this vacuum over the idling jet only and have now arrived at a point where we believe we have this situation in hand.

Fumes from the breather pipe can be taken care of easily and satisfactorily by connecting the pipe on the air-inlet side of the carbureter. If this is done, no further trouble from this need be feared.

Keeping exhaust and manifold joints tight and making sure that no exhaust gas is being blown in from the engine through the dash should hardly be discussed, as these things must obviously be attended to and, if tolerated, indicate very poor maintenance methods.

Avoidance of Obnoxious Exhaust Gases

M. B. MCGORUM[3]:—New York was the first State to ban the introduction of exhaust gases through pipes in the vehicle body for heating. Operators who still use this system are taking serious chances with the wellbeing of their passengers. It is most dangerous if used with an engine in good condition. If the exhaust is laden with oil vapors from an engine in poor condition, some warning is given the passengers, which carbon monoxide will not give.

Fumes that emanate from the crankcase, while acrid and unpleasant, are not so bad as those expelled from the exhaust during deceleration. During this time the cylinders probably are under vacuum even during the firing stroke and these fumes do not go into the crankcase. To control those which do issue from the crankcase, we carry a vent down toward the ground to about the height of the axle clearance, where the relatively small volume of crankcase fumes is dissipated in the air and cannot get back to the body.

I never have thought that oil, grease or dirt on an engine caused gas fumes. For this to be so, the oil would have to be vaporized, and a carbon residue would be left behind. Oil splashed on a hot exhaust pipe would cause fumes, but carbon and not a generally oily condition would be observable on the pipe.

Short tail-pipes are a prolific source of gas that enters the body even when the windows are closed. We had a chassis equipped with such an exhaust terminating near a rear treadle-door. The treadles were of the mechanical-pneumatic type and were perforated. By placing the vehicle in the sunlight, gas could be seen shooting up through them into the body. Therefore we carried the tail pipe to the left rear corner, which cured the trouble without our having to attempt to make the body tight; but, with the windows open, gas still was blown in through them.

Poor carburetion is a major cause of obnoxious exhaust. As Mr. Cumming's report states, mechanics frequently enrich the mixture in an attempt to overcome stalling. One way to minimize this is to use fixed-jet carbureters. The only adjustment then possible is on the air bleed to the idling jet, and load adjustments are not disturbed. Except for short tail-pipes or leaky exhaust lines, exhaust gases will not give much annoyance under major throttle openings if this range of adjustment is even approximately correct. More often than is expected, motor coaches are run on rich mixtures because the choker butterfly is not adjusted to return to correct running position.

In conversation, Mr. Cumming has convinced me of results to be gained in decreasing fume formation by proper accelerator manipulation. While agreeing, I believe that to expect a large group of drivers to handle the throttle ideally from this standpoint year in and year out, with all the other things they have on their minds, would be expecting too much.

The most important item in the list, in my opinion, is that of high manifold vacuum. I place importance on it because, in some measure, we have controlled crankcase fumes, made exhaust lines tight, run them to the discharge point that gives best results and have given much attention to carburetion. None of these things can prevent the cloud of acrid, sickening, white smoke that rolls out of the exhaust during deceleration and pours into windows if they are open. Once it gets into a motorcoach, forced-draft ventilation or other means to clear it out is all very well; but, if means are found to successfully *prevent the formation* of these fumes, we shall hear little more from the public on the question.

Recent developments show gum incrustations on the interior of inlet manifolds to have a place in the picture. Extreme cases have been found where 1½-in. manifolds have been plugged with gasoline gum until they have an opening about the diameter of a cigar. Under heat and mixed with dirt particles, the crust becomes porous and absorbs gasoline. When the throttle is closed, the high vacuum draws this fuel out, enriching the mixture much more than would be caused by the increased flow through the idling jet. Merely shutting off the high vacuum from the carbureter during the overrunning period will not overcome this source of fume formation.

Toronto Commission's Remedial Measures

GEORGE W. WRIGHT[4]:—A brief summary of the various steps taken by the Toronto Transportation Commission to eliminate odors inside its equipment may be of interest.

An investigation of reports submitted by a staff whose duties were to ride all classes of motorcoach and motorbus equipment and report on noise, drafts, fumes and ventilation, resulted in the conclusion that many of the so-called "gas fumes" could be attributed to many sources. To correct these, our engines were first given consideration. All were thoroughly cleaned of oil, grease and road dirt, reconditioned engines were given a coat of enamel before reinstalling in chassis and all chassis parts under the bonnet were thoroughly cleaned and painted. Enam-

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[1] Report by S.A.E. Subcommittee on Motorcoach and Motor-Truck Ventilation, published in the S.A.E. JOURNAL, November, 1932, p. 28. The author, W. J. Cumming, is Chairman of the Committee and general superintendent of the Surface Transportation Corp., New York City.

[2] M.S.A.E.—Superintendent, autobus department, Montreal Tramways Co., Montreal, Canada.

[3] Acting manager of transportation, Virginia Electric & Power Co., Norfolk, Va.

[4] A.S.A.E.—Superintendent, garage department, Toronto Transportation Commission, Toronto, Canada.

eling the engines reduced the labor of cleaning at inspection periods, and clean engines have perceptibly reduced the odors inside of our vehicles.

Our exhaust manifolds and exhaust-pipe connections are now serviced with copper-asbestos gaskets treated with asbestos cement, and manifolds of the slip-joint type are treated with special heat-resisting cement to prevent fumes; but, owing to expansion and contraction of these manifolds, they have to be serviced once a month. Where pocketing of fumes underneath the coaches was found to be caused by the location of the exhaust pipe, the pipes were relocated, in some instances being brought out to the front of the left wheel-housing and in other cases being extended beyond the rear of the body, bent downward and fan-tailed.

Hot-Water Heaters Installed

Some vehicles had considerable flexible armored tubing in connection with the heating system. Such tubing has been replaced with steel tubing and all necessary flexible tubing cemented with asbestos cement, wrapped with asbestos tape and recemented on the outside. All tubing was bracketed to prevent breakage from excessive flexing and vibration. All heater pipes were examined for porosity and renewed if defective. All exhaust-type heaters on our interurban coaches have gradually been replaced with hot-water heaters, and we anticipate that by the end of November all of our interurban coaches will be equipped with a dual hot-water system.

Considerable difficulty was experienced with fumes from the crankcase breather, the odor of burnt oil being very pronounced. Investigation showed an excessive quantity of sludge in the crankcase oil. Our new specification for oil has reduced the quantity of sludge, and a reduction of oil change from 2500 miles to every 1250 miles has materially diminished annoyances from this source.

Vehicles are carefully checked for excessive oil consumption and smoke and, when these are found, are put over a pit for inspection of the valves, sleeves and rings. This procedure reduced the crankcase odors and helped to keep the interior of the coach free from annoying fumes.

Carbureters have been carefully serviced to prevent gas odors. Temperature tests are applied to our thermostatically controlled flapper valves, and these are adjusted after every 5000 miles. All equipment is given a road test under normal operating conditions and adjustments are made to give maximum efficiency, better combustion, less odor and greater mileage.

Coasting economizers have been tried and our experiments to date show that considerable overloading of carbureters is done away with at idling or coasting speeds and that the resultant fumes and smoke from incomplete combustion of gases are minimized.

Careful attention has been given to the dash and toe-plate boards, to prevent engine and road fumes from entering the coach, the openings for clutch, brake and accelerator pedals being protected with rubber and holes for hot-water tubing and other openings fitted with asbestos gaskets.

Floors have been insulated to prevent fumes and cold air from entering the bodies, which were examined for openings through which these might penetrate and were serviced accordingly. All windows have been reserviced with new felts to prevent drafts and aid the functioning of the side and roof ventilators, which are serviced and cleaned after every 5000 miles.

Fans have been tried at the front and the rear of coaches to prevent pocketing of air and to help to keep it in circulation. Hood louvers and various modifications of cowl ventilators have been tried with more or less success, and the equipment is continually sprayed with deodorant.

All tests or experiments made to date, or combinations of them, have been successful to some extent in keeping exhaust gases, engine fumes and road odors from the inside of our equipment. Improvements that have been made were carried out as simply and effectively as possible to do away with the cost of continual maintenance. While our troubles from odors that are so inherent in motorcoach and motorbus bodies have been considerably minimized, they are by no means extinct.

Difficulties of Truck-Cab Ventilation

T. C. SMITH[5]:—While considerable effort has been, and is now being, directed toward proper ventilation of motorcoach bodies, very little thought seems to be given to the ventilation of truck cabs. This is an important problem because the volume of air in the cab is comparatively small and the cab, being located next to the engine, may become so hot in the summer that the condition is almost unbearable for the operator.

The practice of hinging the cab windshields slightly above the center, with the idea that they can be pushed out at the bottom and thus permit air to enter the cab from the bottom and the top of the windshield, is not satisfactory because a heavy draft of air enters the cab over the top of the windshield, circulates under the roof and blows down the back of the driver's neck. While this is occurring, the dead-air space about his legs and feet may be too hot for comfort. Therefore, cabs should not be ventilated over the top of the windshield.

Ventilators have been used to some extent on the side of the cowl. When these take the air from the forward side, the air is often driven back by the fan through the sides of the hood and is not only hot but carries fumes from the crankcase breather. Where these side cowl-openings are made toward the rear, the idea is to have them exhaust the air from the dead-air space at the bottom of the cab. This usually is not effective. The most satisfactory results are obtained by methods that introduce any desired volume of good fresh air into the dead-air space at the front and bottom of the cab. This can best be accomplished by introducing air from the top of the cowl.

Should Ventilate Trucks from Front

J. M. ORR[6]:—The coming of cooler weather formerly brought many exhaust-gas complaints, particularly from crews riding in truck bodies having open rear ends and tarpaulin covers. The men fold or roll the tarpaulin forward to get all the air possible in warm weather but pull the covers up tight with the first threat of winter.

By a process of elimination we came to the conclusion that the odorous gases complained of were discharged at such a point that they entered the low-pressure area in the rear of the vehicle and were drawn into the body. Some of the vehicles complained of were equipped with seamless tail-pipes extending to the extreme rear of the body, which has helped in many cases. After much experimenting, we found that virtually all of the complaints could be eliminated by locating the tail pipes so that they discharge fairly close to the front of a rear wheel, the violently agitated area around the moving wheel serving to dissipate the exhaust gases so that none, or only a negligible portion, of them enter the rear low-pressure area. This assumes good mechanical condition, a clean engine, good carbureter adjustment and a tight exhaust system.

Prior to our adoption of this method, smokestack exhausts, which are now prohibited by city ordinance in Pittsburgh, were used to advantage in some cases, particularly on trucks engaged in underground cable work, where the conventional tail pipe discharged too close to manholes or vaults in which men were working. While the smokestacks removed this hazard, and also the one to the men riding on the truck upon which they were installed, the gases were most objectionable to persons in following vehicles. We have had cases of men being burned on stack exhausts attached to the side of the cab, even though seemingly adequately insulated.

In the design of new truck bodies, adequate ventilation should be assured from the front, through the use of open areas, air tubes or ventilators, with some means for increasing or decreasing the volume of air introduced but not to allow full closure. The flow of air thus introduced minimizes the difference in air pressure between the front and the rear of the body.

In motorcoach operation, good ventilation and the elimination of offensive odors has a definite commercial value and effect. The riding habit is difficult to cultivate if passengers are compelled to ride in fume-laden vehicles.

I believe that this matter of obnoxious exhaust gases is deserving of greater attention than it has been receiving. We are attempting to mitigate the effect, whereas much more might be gained by trying to remove the cause. Its evils contribute to the public's present antagonism toward larger vehicles on the roads and highways. The Society might well consider sponsoring research on this subject by a qualified agency.

[5]M.S.A.E.—Engineer, automotive equipment and construction apparatus, American Telephone & Telegraph Co., New York City.

[6]M.S.A.E.—General manager, Equitable Auto Co., Pittsburgh.

News of the Sections

Triple-Activity Meeting at Detroit

Weight-Saving Cast Iron, Glass and Newspaper Airplanes Interest Three Groups

MORE than 300 members and guests of the Detroit Section were present at the monthly meeting of the Detroit Section at the Book-Cadillac on Nov. 7, which, as in October, was divided into three simultaneous activity sessions. These followed a general members' dinner attended by 235, at which entertainment was provided and which was followed by a business meeting. The meeting then split up into technical sessions of the Passenger-Car Activity, attended by about 250; the Body Activity, at which about 80 were present, and the Aeronautic Activity, attended by 25.

At the general meeting General Manager John A. C. Warner spoke at some length concerning the progress the Society is making both nationally and among the Sections. He said that its activities are showing remarkable progress in this time of depression and called attention to a number of events and features of Society work that have shown better performance than before the depression. One of these was the biggest meeting ever held by the Society, which was the Annual Meeting in Detroit last January. Another outstanding feature has been the new program of group meetings so successfully introduced by the Detroit Section. He asked that every member consider it his privilege and duty to make the Society a very friendly organization, in which every member can exert a helpful influence toward his fellow members, such as encouraging the employment of members who have been temporarily laid off by organizations that are, or soon will be, in need of the best in the way of engineering talent.

Electric-Furnace Cast Iron

At the Passenger-Car session electric-furnace cast iron was discussed from the foundry viewpoint in a paper by A. E. Hageboeck and W. R. Jennings, of the Frank Foundries Corp., of Moline, Ill.; from the designing engineer's viewpoint by H. T. Woolson, of the Chrysler Corp., and from the metallurgist's viewpoint by L. A. Danse, of the Cadillac Motor Car Co.

In presenting the first paper, Mr. Hageboeck dealt with the application of cast iron made by the electric-furnace process to automobiles, motor-trucks and motorcoaches. He reviewed the progress made in metallurgical control of cupola operations in the last 15 or 20 years and then told of the possibilities of the electric-furnace process as supplementary to the cupola method. The electric furnace, with its greater flexibility, higher temperature and close control, he said, opens up to the gray-iron foundryman a field that would not otherwise be possible. Almost any composition desired can be made by the process and, as small quantities can be produced economically, the designing engineer can obtain almost any kind of a mixture and pour it in regular-production molds for sample purposes to try out new ideas. The process also offers an unlimited opportunity to the metallurgist to prove or disprove theories quickly and inexpensively. Service parts that are out of regular production can be made in small quantities and, because of uniformity or other qualities of electric-furnace cast iron, it can be specified for new products and new uses.

Mr. Jennings followed his co-author with a description of a test now being considered for determining the point of optimum superheat for lifting iron from a static to a dynamic condition, with tensile strength of alloyed cast iron of 80,000 lb. per sq. in. and of heat-treated iron of 100,000 lb. per sq. in. When this field is entered, increased temperature becomes necessary for consistent results, and a series of tests is being run to discover approximately the temperature at which breakdown of the carbon nucleus occurs. The speaker showed a cylinder casting 16½ in. long poured at 2950 deg. fahr. through a 5/16-in.-diameter gate and having a Brinnell hardness ranging from 241 to 255 across the face.

The electric furnace, said Mr. Jennings, offers a non-oxidizing and non-contaminating method of melting iron at any desired temperature and allows iron to become high-brow and choosy.

Possibilities of Weight Saving

Mr. Woolson pointed out that designers are continually trying to make 1 lb. do the work of 2 lb., but in this age of alloy steels are prone to underestimate the important possibilities of cast iron in automotive engineering. He suggested that the Standards Committee of the Society consider the advisability of establishing standards for gray-iron castings. The recent development of improved methods of handling molten metal for casting, he said, lends these methods to the obtaining of a much-desired uniformity of castings and physical properties. Some of the readily obtainable properties of electric-furnace iron are strength approximately double that of ordinary cast iron, increased wear resistance, reduced growth characteristics, heat resistance and corrosion resistance.

Mr. Danse dealt mainly with both steel and electric-furnace cast iron brake-drums and their deficiencies and showed with lantern slides the smaller extent of scoring of the iron drums. He also discussed the crystalline structure of different brake materials and how it affects the properties. Considerable experimental work has been done with electric-furnace cast iron, which, he said in conclusion, affords a means of flexibility and control for the purpose, but more work remains to be done along this line.

Motion Pictures of Glass Making

In the Body Activity session, glass making was reviewed by R. A. Miller, technical sales engineer of the Pittsburgh Plate Glass Co., from its discovery in 4000 B. C. to the present. Like most processes and materials of great benefit to mankind, glass making was discovered by accident, he told his audience, but until a few centuries ago glass bore little resemblance to the clear, flat product of today. Motion pictures were shown depicting the casting of plate glass, with the molten glass flowing from the furnace at a temperature of 2600 deg. fahr. and forming a continuous ribbon 7 ft. wide and 900 ft. long and then being laid on the grinding table by several men doing the fox trot on top of the large glass plate.

Case-hardened glass was demonstrated by the speaker by dropping a few lights of this very tough glass on the floor without breaking them. Small samples were passed around for inspection. While made to withstand large pressures and having a very tough outer surface, this glass almost vanishes upon being broken.

The new anti-glare duplate safety glass, said Mr. Miller, not only offers safety from flying fragments in case of an accident but does not discolor or develop edge separation during the normal life of an automobile in any climate. It also filters out a considerable portion of the light rays that cause eye strain and fatigue, the amber-green tint stopping the rays at the red end of the spectrum but allowing the passage of virtually all of the rays from orange to blue, which are the best light for visibility. Because of its reduction of glare, this new safety glass, which will be offered in many cars next year, will go far toward making night driving more pleasant, according to the speaker.

Newspapers Distributed by Airplane

Although newspaper publishers only recently began using airplanes for the distribution of newspapers, enough progress has been made to indicate that they will ultimately be one of the chief patrons of airlines and one of the larger owners of business planes, said James V. Piersol, aviation editor of the *Detroit News*, at the Aeronautic Activity session. Seven daily-paper publishers now own planes, 350 are feeling their way into air transportation by the use of commercial airlines and 31 are regularly chartering airplanes for covering big stories. Three major requirements of an airplane for

newspaper reporting work were given as (*a*) speed, (*b*) a carrying capacity of 500 lb. exclusive of pilot and fuel and (*c*) freedom from obstacles to aerial photography. The *Detroit News* first bought a Lockheed monoplane and 18 months later bought an Autogiro. Both surpassed expectations from the viewpoints of utility and dependability, and 1606 flights on 514 assignments have been made. Many interesting details of work with these planes were told by Mr. Piersol.

The paper was discussed by Prof. Peter Altman, of the University of Detroit; Herbert G. Winter, of the Briggs Mfg. Co., and Walter C. Keys.

Aircraft Radio Explained at Meeting of Washington Section

COMMISSIONER W. D. L. STARBUCK, of the Federal Radio Commission, was one of the speakers to a group of approximately 50 members and guests at the Nov. 16 meeting of the Washington Section, held at the Racquet Club. His subject was Radio as a Factor in Aeronautic Safety. Frederick R. Neely, chief of the aeronautic information division of the Aeronautics Branch of the Department of Commerce, spoke on Analysis of Aviation Accidents. William P. MacCracken, Jr., Chairman of the Section, presided. An excellent chicken dinner was served at 7:00 p.m. and the meeting started at 8:00 p.m.

Only certain wave frequencies are suitable for aircraft use, explained Commissioner Starbuck, hence the Federal Radio Commission selects or allocates the available frequencies. The 3000 band is available anywhere in the United States for the itinerant flyer, while commercial planes have other bands. Canada cooperates with the United States in giving a uniform service everywhere north of the Mexican border. The same frequencies are not suitable for use both day and night. Change of frequency often is made while planes are in the air. The pilot changes over by throwing a switch at a specified time, all land stations and planes changing over in less than 1 min.

The time factor in aviation communication is very important, as a falling plane goes down very quickly. In marine radio service some time usually is available while a boat is filling with water.

The Aeronautical Radio Corp. is a non-profit organization of all 132 aeronautical stations, the radio service of which has developed in the last three years.

Regarding accident prevention, Mr. Neely told the members that the Aircraft Accident Board consists of two pilots, a flight surgeon, an aeronautic engineer, a lawyer and a statistician. About 52 per cent of all accidents are attributable to the pilots rather than the airplane, weather, engine or other causes.

The board does not give out detailed information but classifies accidents by percentages. All such information is published in bulletins issued every six months and can be obtained free of charge. Detailed information gathered by the board is supplied on a strictly confidential basis, which enables manufacturers and operators to obtain many facts that would otherwise be withheld.

The attitude of airline operators toward publicity of accidents has changed in the last few years, said Mr. Neely, and, instead of trying to withhold information from the press, newspaper men are called in and given all the facts, thus creating a sympathetic press.

In scheduled air-transport services for the four-year period 1928 to 1931, the accidents totaled 440, said Mr. Neely. The greatest cause of these was the weather, approximately 27 per cent. The next greatest cause was engine trouble, and the next, airplane structure failure. The last half of 1930 had only three fatal accidents in transport service. This gave more than 25,000,000 passenger-miles per passenger fatality. Since then the ratio has dropped but is still better than in previous periods.

Met Section Holds Diesel Meeting on Motor Ship

AFTERNOON inspection of the new Diesel-driven Grace Line ship Santa Maria, at her Brooklyn pier, a members' dinner aboard and a technical session on Diesel Marine Engines were the attractions that drew 125 members and their guests to the Nov. 17 meeting of the Metropolitan Section.

The speaker of the evening was Louis R. Ford, editor of *Motorship* and associate professor of mechanical engineering at Columbia University. He spoke of the reasons that are hindering the wider adoption of Diesel engines in large vessels built in this Country and said that the oft-repeated assertion that the Diesel engine is inherently unreliable is without basis in fact. As an interesting example, he mentioned that when the steam-turbine-driven auxiliaries on the new Italian liner Rex failed, the Diesel auxiliary set operated without difficulty. He also said that while the vessel was tied up in New York City undergoing repairs, two of the Diesel-electric boats of Electric Ferries were moored alongside acting as floating powerhouses until the small turbines were put back in operation. Some examples of the more common major derangements of Diesels and the methods of repairing them were cited.

In the discussion that followed the paper, Mr. Edwards, superintending engineer of the Grace Line, told some of his experiences with the 8000-hp. Sulzer engines that drive the vessel and the changes he had made to correct certain major difficulties. One of the most pertinent facts mentioned was that replacement parts can be obtained in this Country which are better than the foreign-made ones and at less cost.

Harte Cooke, president of the McIntosh & Seymour Corp., spoke briefly, as did Julius Kuttner, of *Diesel Power*.

Mr. Kuttner stated that the automotive Diesel must not be judged by the type of repairs needed by large marine engines, as heat stresses in the two types are entirely different. As for engine cost, he pointed out that automotive engineers are beginning to realize that a basic design can be used equally well for either a gasoline or a Diesel engine, the only necessary change being in the combustion-chamber and cylinder-head, and that when this change is made the present high cost of the small Diesel can be considerably lessened.

Baltimoreans Discuss Traffic Safety

HIGHWAY traffic and safety were discussed from various aspects by five speakers before a gathering of 57 members of the Baltimore Section and their guests on Nov. 17 at the Emerson Hotel. Addresses on the subject were made by H. M. Lucius, secretary of the Automobile Club of Maryland; Inspector George E. Lurz, of the Baltimore City Police Department; Preston D. Callum, chairman of the traffic committee of the Baltimore Safety Council; David Beecroft, manager of the New York City office of the Bendix Aviation Corp., and Maxwell Halsey, of the National Bureau of Casualty and Surety Underwriters.

Daytonians Told about Refrigeration

RECENT Developments in Mechanical Refrigeration was the subject of a talk given at the Nov. 17 meeting of the Dayton Section at the Engineers Club. The address, which followed a members' dinner, was made by E. W. Newill, vice-president of the Frigidaire Corp., who demonstrated some of the recent mechanical developments by working models of popular makes of refrigerators and described instrumentation for studying refrigerator performance.

Tractor Economics and Tires Interest Big Milwaukee Gathering

MILWAUKEE Section's November meeting on the 2nd was out of the ordinary in a number of respects and was one of the best it has ever held. Being a tractor meeting, it was held at the clubhouse of the Allis-Chalmers Mfg. Co. in West Allis, adjoining Milwaukee. Open house at the plant was held from 7 to 8 p. m. for inspection of tractor and laboratory exhibits. Two excellent papers were presented, the first on Some Economic Aspects of General-Purpose Tractors, by F. N. G. Kranick, of the J. I. Case Co., and the other on Farm Tractors and Low-Pressure Air Tires, by Walter F. Strehlow, of the Allis-Chalmers company. Nearly 200 attended the meeting, guests to the number of 130 outnumbering members by more than 2 to 1. Instead of the customary members' dinner, a complimentary buffet supper was served after the close of the discussion at 10:35 p.m., when a rising vote of thanks was tendered to the speakers and to the Allis-Chalmers company for use of the clubhouse and the laboratory equipment and motion pictures shown at the technical session.

Preceding the presentation of the

papers, the members elected Section representatives to the Nominating Committee of the Society and also elected P. C. Ritchie, of the Waukesha Motor Co., as its representative on the Sections Committee for 1933.

Design Must Be Commercialized

The burden of Mr. Kranick's presentation was that tractor engineers must translate the wants and needs of the tractor user into useful and profitable designs. The first essential toward designing a successful unit, he said, is a study of the requirements of the general-purpose tractor. The major requirements are that it must be (*a*) of benefit to the purchaser and serve as a means to profit and (*b*) sold at a price that he can pay.

Much radical design frequently gets into the tractor picture, asserted Mr. Kranick. It may be part of the development process of shaping the tractor into a commercially practical machine and usually results in complicated mechanism. Good engineering simplifies, and the better it is the simpler the product is. Tractors are now on the market that are exceedingly simple by comparison with earlier designs and will perform best and survive in the end.

Close cooperation and harmony between the engineering and sales organizations is necessary to success, declared the speaker, who made a plea for more conservative common-sense design that will make for public acceptance of the product, reduce sales expense and represent more value to the man who buys it.

Machinery that goes with the tractor should be diligently studied, as the tractor is useful only as it accomplishes work by means of the equipment that goes with it. Further, tradition is a great handicap, according to the speaker. Purchasers' habits and customs frequently have to be changed, but to try to make this change too quickly sometimes is dangerous. The engineer who can bring out a mechanism that least disturbs tradition will find a quick acceptance for it which will mean a large market if it is a means to a useful end.

Low-Pressure-Tire Tests

Charts and data from tests of low-pressure pneumatic tires on farm tractors were presented by Mr. Strehlow, who mentioned in his introductory remarks that the latest available census showed that considerably more than 1,000,000 tractors are in operation on American farms. Solid and high-pressure pneumatic tires used on industrial tractors did not prove successful on farm tractors. A low-pressure airplane tire with thickened side walls and carrying a pressure of 12 lb. has been tested on some Allis-Chalmers farm tractors. It gives an average ground pressure that compares with a pressure of from 6 to 8 lb. obtained with crawler-track tractors. These tires are designated as 20 x 10 — 46 in. Tests showed that at any tractor speed the tires used only one-half the power required by steel wheels with lugs and that smaller engines can be used in tractors and still give the same results at the drawbar as with steel wheels, according to Mr. Strehlow, resulting in a considerable saving in fuel. Or, if the same engine is used, operations can be performed at higher speed.

To take full advantage of these tires, said Mr. Strehlow, the tractor should have at least four forward speeds with a high speed of 10 to 20 m.p.h. for use on the road. The tractor company does not believe that the tires are a cure-all but does believe that they have important advantages and that they will be used on a large number of tractors in the future.

Tractor Requirements and Performance

TWO papers were presented at the tractor meeting of the Northern California Section at the Elks Club in San Francisco on Nov. 9 following a members' dinner. An instrument for the quick and accurate measurement of tractor field performance was described by J. M. Davies, chief experimental engineer of the Caterpillar Tractor Co., and the basic demand requirements for farm tractors were discussed by Prof. B. D. Moses, of the University of California Agricultural Station.

Sixty members and guests of the Secton were present and at a brief business session elected a representative and an alternate to serve on the Nominating Committee of the Society and also elected John M. Evans as the Section's representative on the Sections Committee.

The December meeting, to be held on the 13th at the Athens Athletic Club in Oakland, will be devoted to Diesel engines and fuels.

High-Compression Engine Factors Debated at Los Angeles

IN an informal talk from notes. which was illustrated with free-hand drawings on the blackboard, Ethelbert Favary spoke on High-Compression Engines before a gathering of 194 members and their guests at the Nov. 4 meeting of the Southern California Section at the Frank Wiggins Trade School in Los Angeles. The technical session followed a members' dinner attended by 136.

Mr. Favary discussed the reasons for high engine efficiency, detonation and its cause and cure, the cause of rough engine-operation and the remedy, the advantages and disadvantages of turbulence and combustion-pressure rise. He also took up the subject of the design of high-compression-engine heads and told of the results of researches in this field by H. R. Ricardo and W. A. Whatmough in England and Robert N. Janeway in this Country to ascertain the best engine-head design to give the maximum miles per gallon.

In the discussion on the talk, Carl Abell, of the Ethyl Gasoline Corp., told of the work that his company is doing to promote the use of high-compression engines; W. B. Birren, of the Wright Aeronautical Corp., discussed cylinder-head design in airplane engines and illustrated his points with blackboard drawings, and Section Chairman C. F. Lienesch also brought out some interesting points.

Deputy Chief Davis, of the Los Angeles Police Department, who was introduced by the Chairman, gave a short talk on highway-traffic conditions in the city.

New Englanders Told of Motorcoach and Truck Developments

MOTORCOACHES and motor-trucks of the latest design, including the new Mack Stratotherm, were exhibited at the Mack truck plant in Boston and described by A. F. Masury, of the International Motor Co., at the Nov. 9 meeting of the New England Section. About 80 members and guests were present at the dinner, held in the Walker Memorial building at the Massachusetts Institute of Technology in Cambridge, Mass., and were transported in three motorcoaches of the latest type, one with power steering, to the technical session at the truck plant. Approximately 150 attended the session.

At the December meeting of the Section it is hoped to have a talk on down-draft carbureters.

Why Airships? Answered by Dr. Arnstein at Cleveland Meeting

THAT a regular, scheduled airship transatlantic passenger and freight service has been in operation for the last year probably is not realized by a great many persons, even among a technically educated group, Dr. Karl Arnstein, vice-president of the Goodyear-Zeppelin Corp., told a joint meeting of the Cleveland Section of the S.A.E. and the Cleveland Chapter of the American Society for Steel Treating on Nov. 11.

The speaker presented an extensive paper entitled Why Airships?, illustrated with motion pictures, before a record Section gathering of 425 members of the two societies, which was held at the Cleveland Club following a dinner attended by 125.

Impediments to Commercial Development

Dr. Arnstein spoke of the successful solving of the many technical problems involved in the plans of Count Zeppelin to conquer the air with large rigid airships and then the sudden diversion of public interest to the airplane as a result of the practical development of the light-weight internal-combustion engine. Again, when continued research and development had once more brought the possibilities of the airship to the public notice all over the world, financial backers who were preparing to assist the airship in the commercial field found themselvs unable to go forward with the plans, which have therefore remained dormant.

The speaker then discussd the fundamental reasons for the airship proving successful in the field of transatlantic flights where other means of high-speed transportation have been unable to supply acceptable performance, and reviewed the history of the development of the airship, illustrating the progress with lantern slides. He spoke of the

military uses of lighter-than-air craft, including the U. S. S. Akron, their utility for polar exploration and, finally, their commercial uses.

The aerodynamic characteristics and mechanical features of airships and the interdependence of various means of transport in the matter of mechanical development were pointed out and the long-distance flights of various ships were recalled. Data were given on the 232 voyages made up to Jan. 1, 1932, by the Graf Zeppelin.

In conclusion, prior to the first public showing in America of motion pictures of the early stages of Zeppelin development, and films depicting the building of the Los Angeles in Germany and the Akron and Macon in America, Dr. Arnstein discussed the coordinate fields of the airplane, the airship and the steamship. The field for the large airship, he said, is clearly defined and the only obstacle is the lack of legal precedent and Congressional statutes authorizing this means of transportation and assisting it while passenger revenue is being developed. When such legislation is enacted, the airship builders will be in a position to interest financial groups in this new and coming phase of transportation.

Hydrofoil and Hydroplane Boats Considered by "Met" Marine Division

TWO papers on advanced motorboat design were presented at the first meeting of the season of the Marine Division of the Metropolitan Section, which was held on Nov. 3 at the Hotel New Yorker following a members' dinner at 6:15 p. m. The dinner was attended by 37 and the technical session by 75 members and their guests. William Edgar John, Vice-Chairman of the Section for marine engineering, presided.

Hydrofoils for Higher Speed

The first paper, entitled A New High-Speed Watercraft, was given by Dr. Oskar G. Tietjens, consulting engineer of the Westinghouse Electric & Mfg. Co., who spoke of the general trend of all transportation toward higher speeds and said that in most cases the increase in speed should be credited to the improvement of motive power and the increase in engine speed. In small and medum-size motorboats, however, the higher speed is attributable to the change in hull design from the displacement type to the hydroplane type.

With this development, we have stopped half way, in Dr. Tietjen's opinion, as, instead of moving an inclined plane, namely, the bottom of the boat, over the water surface so as to support a large part of the weight of the boat, moving so-called hydrofoils through the water and using their lifting forces to counterbalance the weight of the boat is much more efficient.

Even with the adoption of the hydroplane type, a boat built for excessive speed is less economical than any other means of high-speed transportation, the speaker said. If one were to build a boat having a top speed of 100 m.p.h., from 3000 to 4000 hp. would be necessary.

Dr. Tietjens presented many slides and some motion pictures showing the testing of the type of hydrofoil that he believes will greatly increase the speed of high-speed motorboats and also exhibited the test model with which the original experiments were made.

The difference in performance of a plane gliding over the water and a hydrofoil moving through the water was discussed, the pressure distribution for a planing surface and a hydrofoil being shown by a lantern slide. Results of tests to determine the lift-drag ratio for the new type of hydrofoil were given in a curve shown by a slide. Because water is about 800 times as dense as air, the speaker said, the hydrofoils need be only 1/800th of the size of airplane wings for the same weight and size of craft.

A New Pontoon Hydroplane

The next speaker, Thomas A. Edison Lake, described a new type of pontoon-hydroplane boat. His paper dealt with the question of safety and comfort in combination with speed. He stated that, in the pontoon-hydroplane type of boat, he has provided a strong but light body that does not rest directly upon the water. It does not have to be watertight nor stand the shock or pounding of water upon it. In purpose it is merely a girder or framework supporting any desired type of body for the passengers, aerodynamically streamlined and forming no part of the flotation system itself. The body and the flotation system being divorced from each other means that each can be designed for its own purpose and the maximum efficiency in streamlining of both be achieved.

Arms extending from each side of the forward part of the framework connect with the two forward pontoon-hydroplanes, which really are small watertight hulls that are streamlined and have stepped planing surfaces. The two forward pontoons are connected to the ends of the side arms so that they can be tilted about a longitudinal axis as well as a lateral axis. Depending upon the weight of the body supported by them, they can be set for maximum planing efficiency. Oleo shock-absorbers form a part of the float connections and automatically adjust themselves to the changing surface of the water. They provide the same easy riding for a boat that is provided for a motor-car by balloon tires, springs and snubbers, according to Mr. Lake.

Located in the rear is a third float similar to the two forward and having a resilient shock-absorbing attachment. This is pivoted so that it can be turned to either side like a rudder, and when so turned, the stern of the craft tends to swing around in the direction to which the rear float is pointed. The turning of the rear float is simultaneous with the banking of the other two, so that they give a lateral thrust to offset skidding and force the bow to one side while at the same time checking the forward momentum at this point.

Mr. Lake showed very interesting motion pictures of the pontoon-hydroplane in action.

Among the naval architects and boat and engine builders who discussed both of these papers from the floor were George F. Crouch, P. L. Rhodes, Alfred E. Luders, F. K. Lord and W. E. John.

Fleet Costs and Employe-Owned Cars Considered at Pittsburgh

MOTOR-VEHICLE fleet operators have now come to the stage where they must hunt for penny savings, whereas in former years they were urged and advised how to avoid dollar waste, said F. K. Glynn, of the American Telephone & Telegraph Co., in a paper presented at the Nov. 10 meeting of the Pittsburgh Section. The other speaker of the evening was John M. Orr, manager of the Equitable Auto Co., who delivered the paper on The Use of Employe-Owned Cars in Business which he gave at the Transportation Meeting of the Society in Toronto in October.

The meeting was held in the Fort Pitt Hotel. Owing to two changes of date and the short notice of the final date selected, the attendance of 40 at the dinner and 55 at the technical session was not so good a representation of members as the quality of the papers deserved.

At a brief business meeting, the Section elected representatives to the Nominating Committee of the Society and elected Fred Heisley to represent the Section on the Sections Committee of the Society next year.

Must Consider Records Seriously

In his address, Mr. Glynn said that we must not overlook present possibilities of savings in the cost of operation and maintenance because they are smaller than formerly, and that in the exchange of ideas made possible by S.A.E. meetings we will find the fertility of a fresh point of view. He presented a neat mathematical formula for easily calculating costs to a definite target. He said that he was reporting ideas gathered from other fleet operators, which is one of the practical advantages of membership in the Society.

That a wide diversity of practice exists in the operation of large fleets and in record keeping was brought out in discussion of the paper by operators representing many millions of miles of transportation per year. Mr. Glynn said that the time has come to give records serious consideration; complete records cost money, but much can be learned from records of only gasoline consumption and vehicle repairs, spot-checking when necessary, and using work-performed records of cars and trucks requiring special attention.

Fleet operators who contract for tires on the mileage-purchase plan reported the keeping of accurate tire records, but in other cases tire records were found to be open to question, as tire men depend mainly upon their ability to tell what service a tire has given merely by examining it.

Oil Changing Stirs Up Debate

Mention of lubricating oil stirred up lively debate. Mr. Glynn referred to a test in which a truck had been driven

63,000 miles in a period of several years without a change of oil and that it had miked up as showing less wear than another truck which had done approximately the same work and in which the oil had been changed every 500 miles. Where the oil is not changed, the quality of the oil and its viscosity must be right, but a quantity of sludge in the bottom of the crankcase does no harm.

Mr. Yule, of the Ward Baking Co., stated that his company changed oil regularly at 500-mile intervals because most of the trucks needed it. Arthur Tiel, of Exhibitors Service, said that he changed oil only at long intervals in the summer, when little water-vapor condensation occurs. The need for oil changing is affected also by whether a truck is operated both day and night or is stored over night in a cold garage.

Need for an oil change is due only to contamination of the oil, asserted Fenley Hunter, lubricating engineer of the Gulf Refining Co., who said that moisture in the crankcase, precipitated by atmospheric conditions, is an important factor in the formation of sludge, an accumulation of which results in a hotter crankcase and increased liability of bearing failure. With oil costing not more than 2 per cent of total operating expense, he asked, is it worth while to run the risk of bearing failure by not changing oil?

Employe-Owned Cars Discussed

Discussion on Mr. Orr's paper brought out the point that the use of employe-owned cars in addition to company-owned cars gives greater flexibility in both use and types of car, especially when some are used in widely scattered locations and cannot conveniently be returned at night to a central garage.

Divorcing of employes from the idea that size of a car is a measure of the importance of a man's job has been found desirable, according to one speaker. When employes in general realize that the company-owned car is a means to an end, standardization on cars of low operating cost is possible. Executives in positions of major supervision may be allowed to choose their own make of car under an equitable arrangement for employe ownership. Such cars are kept in as good condition as company-owned cars, because usually they are more expensive and the owners take special pride in keeping them looking and operating well.

Others who took part in the discussion on the papers were B. H. Eaton, of the Bell Telephone Co.; Charles F. Kells, of the West Penn Power Co.; Joseph Harvey, of the Pittsburgh Motor Coach Co., and Mr. Lancaster, of the Equitable Auto Co.

Tractor-Fuel Classification and Superchargers Considered at Chicago

APPARENTLY no relationship of distillation, gravity and antiknock qualities exists among kerosenes is the conclusion drawn from tests recently made, K. T. Winslow, of the Waukesha Motor Co., told 125 members and guests of the Chicago Section at its meeting in the Hotel Sherman on Nov. 1. He presented a paper entitled Tentative Classifications of Low-Volatility Tractor Fuels, which was accompanied by a number of lantern slides of charts.

Louis Schwitzer, of the Schwitzer-Cummins Co., gave a talk on the use of superchargers on passenger-car and commercial-vehicle gasoline engines, and also discussed at length the Packard V-type engines equipped with superchargers as used in Gar Wood's Miss America X. The talk was supplemented with some interesting motion pictures of the Harmsworth Cup race.

At a brief business meeting the Section elected Robert E. Wilkin to represent it on the Sections Committee of the Society in 1933.

Tests Made with C. F. R. Engine

Because the classifying or grading of kerosenes and other tractor fuels of low volatility is becoming of increasing importance, said Mr. Winslow in presenting his paper, a tentative technique was worked out in the laboratory of the Waukesha company as a step toward classifying these fuels in octane numbers or the lack of octane numbers, as the case might be. If a not-too-fine classification of kerosenes and distillates were nationally recognized and accepted, engines could be designed to burn fuels of either the upper or the lower bracket and the operators could purchase fuels so classified.

Laboratory equipment used in making the tests were a C. F. R. research engine equipped with a modified carbureter; two reference fuels, one a kerosene having the severest detonating quality of the 11 available samples and the other a gasoline used as a secondary reference fuel in making octane ratings of gasoline; 10 kerosene samples obtained at random in and around Milwaukee and 1 furnace oil recommended as a tractor fuel.

Tables of distillation temperatures and time were given for all the fuels tested, and a summation table showed the range in degrees from the initial to the end point in 10-per-cent stages. Octane-number ratings were then obtained. The Baumé and specific gravities and the compression ratio of the engine at incipient knock were listed.

Some of the findings were that the heaviest fuel had the best antiknock quality, although its initial distillation temperature was higher than that of any of the other kerosenes; that another of the better fuels had an initial point very close to that of gasoline and its distillation curve does not approach that of any of the other kerosenes until it reaches the 30-per-cent point, and that one of the worst fuels as regards antiknock quality has a distillation curve that follows the characteristics of the best sample from its 10-per-cent point and of the kerosene reference fuel and a gravity between that of the two better antiknock fuels. The engine operated on the heaviest fuel at a compression ratio of 4.25:1 with no detonation difficulty, and on the lightest fuel at a ratio of 3.65:1.

In conclusion, Mr. Winslow stated that an engine designed to run on the heaviest fuel would prove satisfactory for all the others but considerable power would be wasted if this fuel were used in engines of low enough compression ratio to burn all of the other samples.

Hundreds of samples, instead of 10, should be rated, said the speaker; 5 or 10 laboratories should be working along this line; results should be checked and rechecked, and procedures should be compared and correlated.

Nature of Electric Spark Explained to Northwest Section

THE nature of the electric spark and some practical uses of the Lichtenberg Figures formed the theme of a lecture by Prof. C. Edward Magnusson at the November meeting of the Northwest Section.

The meeting was held on Nov. 4 at the Bergonian Hotel in Seattle with an attendance of about 40, and the usual members' dinner preceded the brain-teasing lecture in which Professor Magnusson, who is in charge of the Engineering Experiment Station and professor of electrical engineering at the University of Washington, delved deeply into a study of the so-called Lichtenberg Figures, the early observations of the famed professor of physics at the University of Göttingen being supplemented by later developments by noted workers in this field of electrical research.

Before presenting the speaker, Section Chairman C. C. Finn displayed some slides of sparks of different kinds cast off from grinding wheels, enabling one to distinguish different alloy steels, up to 1 per cent carbon, and the distinctive nickel and molybdenum sparks. Vanadium, chromium and other elements are determined, said Mr. Finn, by the minute pellets, each having a distinctive appearance. These also were illustrated by slides.

Lichtenberg Figures Furnished Clew

Professor Magnusson, who is widely recognized for his research work into the nature of electric impulses or surges, explained that in 1777 Dr. Lichtenberg observed that under certain conditions an electric discharge produced certain figures on a glass surface around the point of contact of the electrode. One hundred years later, through photography, a method was obtained for visualizing this phenomenon much better than formerly when the figures were made visible by sifting light powder, such as lycopodium, over the glass surface. No practical application had been found of the Lichtenberg Figures until 1924, when J. F. Peters, of the Westinghouse Electric & Mfg. Co., developed the klydonograph. Not until then did engineers realize the importance of the discovery. The General Electric Co.'s surge-voltage recorder, the speaker said, is a modified form of the klydonograph which gives simultaneous recordings of both positive and negative Lichtenberg Figures.

Many slides of both types of figures, showing their variations and characteristics, were thrown on the screen. Effects of the magnetic field on the figures also were illustrated. These

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Guy W. Motz

NEWS of the sudden death of Guy W. Motz, Western advertising representative of the Society, at his home in Shaker Heights, Cleveland, was received on Nov. 5. He passed away as he was being taken to St. Luke's Hospital in Cleveland for an emergency operation on his throat. The passing of Mr. Motz, who had represented the S.A.E. JOURNAL in the North Central States for the last 12 years, will be regretted by his many friends and acquaintances in the industry, who liked him for his great geniality.

Mr. Motz had been an Associate Member of the Society for the last five years. He was born at Akron, Ohio, in 1887, and, after attending the high school there, was graduated from Western Reserve University at Cleveland in 1910 with the degree of bachelor of laws. For the following three years he was in the advertising business for himself and in 1914 became advertising representative of the Class Journal Co., of New York City, publisher of *Automotive Industries* and other business periodicals. Six years later he joined the staff of the Society as advertising representative of THE JOURNAL, in which capacity he was very energetic and successful during the last 12 years.

Ray Prescott Johnson

A PROMINENT Muncie, Ind., industrialist and one of the founders of the Warner Gear Co., Ray Prescott Johnson died Nov. 16 at Phoenix, Ariz., following a heart attack. He was born June 4, 1877, at Bluffton, Ind., and was educated at a military school in Ohio, at Morgan Park Academy in Chicago, at Franklin College and at the University of Chicago, where he received the degree of bachelor of science, in June, 1902.

After being connected with the mining industry for a short time, Mr. Johnson returned to Muncie to become one of the six employes of the newly founded Warner Gear Co., being associated with T. W. Warner and Warren Sample in the enterprise. He was an expert technician, was treasurer of the company and gradually progressed to become its president in 1919.

In addition to being a founder of the Warner Gear Co.—which consolidated with the Borg & Beck Co. and the Marvel Carburetor Co. and thus constituted an important development in automotive manufacture a number of years ago—Mr. Johnson was vice-president of the Borg-Warner Corp., vice-president of the Glasscock Mfg. Co., and president of the Warner Electric Co. He was also a director of the Live Poultry Transit Co., of Chicago.

Mr. Johnson's expert knowledge of automobile mechanics was recognized during the World War, when he was sent by the Secretary of War to France to head a committee in organizing repair shops for the A.E.F. Motor Transport Corps. He spent six months at Tours engaged in that service and traveled in Spain and elsewhere to purchase tools for the repair corps.

Mr. Johnson started his career in the automobile industry at buffing steering-wheel spiders at a wage of $7.50 per week.

Because of ill health, he was induced last fall to go to Phoenix, Ariz., to recover. He was elected to Member grade in the Society in October, 1908.

Arthur Stanley Baldwin

WITH much regret we record the death of Arthur S. Baldwin at his home in Stratford, Conn., on Oct. 24 as a consequence of a nervous breakdown six months previously. Mr. Baldwin was one of the early members of the Society, having been elected to Member grade in June, 1911. For the last two years he had been much interested in the unemployment situation and evolved plans for a 200-ft.-wide transcontinental highway which he submitted to President Hoover several months ago as a project to relieve unemployment and provide a high-speed traffic route.

Born at Joliette, Canada, in 1882, he secured his education at Durham, Canada; Easton and East Village, Conn., and through the International Correspondence Schools.

After serving a regular apprenticeship with the United States Rapid Fire Gun & Powder Co., of Derby, Conn., as machinist and toolmaker, Mr. Baldwin was for 2½ years shop superintendent for the American-British Co., of Stratford, Conn., in charge of tool design and machine work. He then became shop superintendent for the Driggs-Seabury Ordnance Corp. and had charge of 700 men engaged mostly in automobile work. Following five years in this position he was successively general manager of the Newburgh Ice Machine & Engine Co. and the Alberger Pump & Condenser Co., of Newburgh, N. Y., and in 1917 became manager of ordnance for the Poole Engineering & Machine Co., of Baltimore, in which position he continued for five years. Since 1921 he was engaged in private consulting engineering work in Baltimore until 1929, when he moved to Stratford.

Louis Barber

IN the recent death of Louis Barber due to a heart attack, the Society lost a Member who had been in the Society for nearly 12 years, and the National Screw & Mfg. Co., of Cleveland, a mechanical engineer who entered its services in 1900 as a draftsman and again in 1904 as a designing engineer in charge of the engineering department.

Mr. Barber was born at Toronto, Ont., Canada, in 1870 and became a naturalized citizen of the United States in 1900 in Cleveland. He secured his education in mechanical engineering through the course of the International Correspondence Schools. He was a member of the Cleveland Engineering Society and the American Society of Mechanical Engineers. The 10 years from 1890 to 1900 were devoted to wood and metal pattern making for automobile engines. For two years, 1902 to 1904, he served the Brown Hoisting Machinery Co. and the Royal Motor Car Co., both of Cleveland, in the capacity of draftsman.

Moses Edward Cheney

FOLLOWING a long, severe illness, Moses Edward Cheney, chief engineer of the Moto Meter Gauge & Equipment Corp., of Toledo, passed away at Bayport, N. Y., on Sept. 7. His loss is deeply felt, not only by his associates in the company, but by his many friends in the automotive industry.

Mr. Cheney had been connected with the electrical and automotive industries for nearly a quarter of a century. Born at Palouse, Wash., in 1885 and graduated from Washington State University with the degree of bachelor of science in electrical engineering, he entered the service of the General Electric Co., of Schenectady, N. Y., in 1907. From 1909 to 1911 he was instructor in charge of the electrical engineering courses at Bryant College, in Buffalo. For the following eight years he held the position of chief electrical engineer of the Washington Surveying & Rating Bureau, of Seattle.

When the United States entered the World War, Mr. Cheney was commissioned second lieutenant and assigned to the Bureau of Aircraft Production of the Army Air Service Reserve Corps as chief of radio and airplane and spares inspection and chief of the inventory section in the New York district. From 1920 to 1923 he was assistant engineer of the B. G. Corp., of New York City, manufacturers of spark-plugs. In the latter year he established the West Bronx Service Corp., of which he was proprietor until he effected a connection with the National Gauge & Equipment Co., of La Crosse, Wis., in 1925, in charge of experimental and research work. After filling this position for more than four years he joined the Moto Meter Gauge & Equipment Corp. in 1929.

Mr. Cheney was elected to Member grade in the Society in June, 1931.

James Glanding Dailey

AFTER a professional career of more than 30 years as an industrial chemist and metallurgist in this Country and Mexico, James Glanding Dailey passed away at his home in Toledo on Oct. 2. For the last five years he was chief engineer of Defiance Spark Plugs, Inc., of that city.

Mr. Dailey, who was elected to Member grade in the Society in 1929, was born at Philadelphia in 1877 and was graduated from the University of Pennsylvania in 1899 with the degree of bachelor of science, specializing in chemistry. He was a member of the American Institute of Chemical Engineers, American Chemical Society and American Ceramic Society.

For three years after his graduation Mr. Dailey worked as an analytical chemist in this Country and then for 10 years lived in Mexico, where he conducted his own business as a consult-

ing metallurgical engineer. Returning to his native land, he served the Clinchfield Fuel Co., of Spartanburg, S. C., for two years and the Crown Cork & Seal Co., of Baltimore, for three years as chemical engineer. His next connection was with the Tungsten Products Co., of Baltimore, as manager of production of ferrotungsten and ferrochrome in electric furnaces—a war activity. From 1919 to 1921 he was divisional plant manager for the Calco Chemical Co., of Bound Brook, N. J., and, for the succeeding six years, foreign branch manager at Buenos Aires, Argentina, for the National Aniline & Chemical Co., of New York City. In 1928 he was appointed chief engineer of Defiance Spark Plugs, Inc., in charge of design, development and all technical phases of the company's activities.

Henry Collier Smith

INFORMATION has been received from the Quickwork Co., of St. Marys, Ohio, that Henry Collier Smith, owner of the company, passed away in St. Marys on Sept. 16.

Mr. Smith was elected to Associate Member grade in the Society in 1915 and was transferred to Member grade in 1923. He was born at Raleigh, N. C., in 1866 and was engaged in sheet-metal work throughout his long business career. He was successively superintendent for Moore & Munger, of New York City, automobile custom-body builders; in charge of fender, hood and tank production for the Springfield Metal Body Co., of Springfield, Mass.; in charge of body work for the Racine Mfg. Co., of Racine, Wis.; superintendent of the sheet-metal and stamping department of the Cadillac Motor Car Co., of Detroit; estimator on sheet-metal work for Hermann & Grace, of Brooklyn, N. Y.; and proprietor of a sheet-metal works in Brooklyn. About 1917 he founded the Quickwork Co. for the manufacture of heavy metal shears and fender and sheet-metal-working machines of his own design and became president and general manager. In 1927 he purchased other interests in the company and since then was sole owner.

Edward J. Fritton

WITH great regret we record the recent passing of Edward J. Fritton, a member of the Society since 1914 and a member of the Buffalo Section from its organization. Mr. Fritton died at his home in Lockport, N. Y., having retired from business in 1929 because of ill health.

Born at Lockport in 1871, Mr. Fritton received his education in the public schools and through a correspondence school, and much of his business career was devoted to the designing, building and marketing of automobiles and their component parts. He was associated for about 10 years with the Field Force Pump Co., serving as superintendent for nearly one-half of that time. Then for a decade he was connected with the Covert Motor Vehicle Co., of Lockport, of which he was secretary and general manager at the time of joining the Society. He assisted in the manufacture of one of the first automobiles produced by the Covert company and took part in the 1904 Glidden Tour to St. Louis. Next he became general manager of the Friend Mfg. Co., of Lockport, manufacturer of farm spraying-machines, which later removed to Gasport, N. Y., and with which he remained until 1929, becoming vice-president and general manager in 1927.

Earl William BeSaw

A SUDDEN heart attack resulted in the deplored death of Earl William BeSaw, president of the Firestone Tire & Rubber Co. of Canada, of Hamilton, Canada, at his home in Fort Nelson, Canada, on Sept. 5.

Mr. BeSaw, who was elected to Associate grade in the Society in December, 1930, and was a member of the Canadian Section, had been connected with the Firestone company since March, 1911. From the position of salesman in the United States he rose to that of general sales manager in 1920, then became vice-president of the Oldfield Tire Co., a Firestone subsidiary. In December, 1922, he was made vice-president in charge of the Canadian plant, and on Jan. 1, 1930, became president of the Canadian company.

Mr. BeSaw was a native of the tire city of the United States, having been born at Akron, Ohio, in 1886, where he attended the public grammar and high schools. Subsequently he took a correspondence course in commercial law and advertising. All of his business career was devoted to the tire and rubber business of the Firestone company.

Harry A. Scott

WITH great regret we chronicle the sudden death on Sept. 30 of Harry A. Scott, a member of the Society since 1916, at his home in Kalamazoo, Mich., as a consequence of heart trouble.

Mr. Scott was widely known both as an engineer-inventor and as official starter of the Indianapolis Memorial Day 500-mile races and judge in many other automobile and airplane races.

Born at Kalamazoo in 1888 and educated in the high and normal schools, he entered the automobile field at an early age, designing and building experimental engines. He worked for years on the development of the Scott airplane engine. Previously for three years he was mechanical engineer of the Kalamazoo Paper Co. and in 1909 formed the firm of Harry A. Scott & Co., of Kalamazoo, of which he was owner and mechanical expert. In 1929 this was incorporated as the Scott Engineering Co., of which he was elected vice-president and chief engineer, retaining this position until his death. In 1918 he was vice-president and chief engineer of the Wolverine Motor Car Co., of Kalamazoo. In 1924 he invented a motor iceboat. He was very active in local chapters of fraternal organizations.

News of the Sections

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showed bending of both negative and positive streamers, providing means for determining whether electrons, ions or protons are basically the active elements in the formation of the positive as well as the negative figures.

Professor Magnusson emphasized the practical importance of the Lichtenberg Figures in the study of lightning, stating that many records have been made in recent years and much more is now known about the nature of lightning than was known a decade or so ago.

The number of questions asked following the lecture indicated the keen interest taken by automotive men, who are eager to learn all they can about the basic facts or theories of the electric spark in internal-combustion-engine applications.

Hoosiers Warned About Motor Taxes and Legislation

ONE of the Indiana Section's best meetings was that on Nov. 10, at the Indianapolis Athletic Club, which was devoted to consideration of the effect, on highway transportation, of burdensome taxes and present and impending legislation limiting vehicle size, weight and speed. Fifty-seven members and guests were present at the dinner, and the technical session was attended by 90.

A representative of the Section and two alternates were elected to the Nominating Committee of the Society at a brief business meeting, and Charles Merz was elected as a member of the Society's Sections Committee for 1933. An announcement was made that Dr. Kurt, of the United States Rubber Co., will speak on Shimmy and Tramp at the Dec. 8 meeting of the Section.

Excellent talks to the interested members were made by President Scaife and General Manager Warner, of the Society, and by Sam C. Hadden, Secretary of the Indiana Motor Traffic Association. Good discussion on the subjects of the addresses followed their presentation.

President Scaife spoke on the subject of Highway Transportation and the Mounting Automotive Taxes, covering facts that he had given in talks before other Sections and that were published in the S.A.E. JOURNAL for October, 1932, p. 14, under the title, The Burden of Ruinous Taxation.

Mr. Hadden emphasized the responsi-

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Personal Notes of the Members

Having relinquished his position as tool designer and draftsman for the General Aviation Mfg. Corp., of Baltimore, **Franklin J. Behrndt** is now serving the Stromberg Carburetor Co. division of the Bendix Aviation Corp., at South Bend, Ind., as layout inspector.

W. H. C. Carhart is now located with the Youngstown Sheet & Tube Co., in Chicago. Prior to making this connection he was sales engineer in the Chicago office of the Republic Steel Corp.

Virginius E. Clark was recently appointed chief engineer of the General Aviation Corp., of Dundalk, Md. His prior connection was with the American Airplane & Engine Corp., of Farmingdale, N. Y., of which he was vice-president.

Edward W. Dart, formerly engineer in the aircraft department of the Ford Motor Co., at Dearborn, Mich., is now located at Pelham, N. Y., and has not announced his plans for the future.

Roy O. Dent is filling the position of service manager for Davison Bros., of Morgantown, W. Va., an automobile sales and maintenance firm, for which he was night service manager last year. His last position was as engine mechanic with Pennsylvania Airlines, Inc., at the municipal airport at Willock, Pa.

William H. Donaldson, who was regional service manager at Pontiac, Mich., for the General Motors Truck Co., is now service representative in the Detroit zone for the Buick-Olds-Pontiac Sales Co., of Detroit.

A. E. Fawley, sales engineer for the Marlin-Rockwell Corp., has been transferred from Plainville, Conn., to Detroit

William H. Funston, Jr., was recently elected president of the Firestone Tire & Rubber Co. of Canada, Ltd., of Hamilton, Canada. He previously held the position of general manager of the mechanical rubber goods division of the Firestone company at Akron, Ohio.

Edward M. Gureasko, formerly a mechanic with the Curtiss Aeroplane & Motor Co., of Buffalo, is now located with the Metropolitan Aircraft Co., at Floyd Bennett Field, Brooklyn, N. Y.

Elbert J. Hall, formerly vice-president of the DeVaux-Hall Motor Car Co., of Grand Rapids, Mich., has retired from all active business and is now residing in Paris, France.

Herbert H. Henchel has been appointed manager of national fleet sales for the General Motors Truck Co., of Pontiac, Mich. He previously occupied the position of president and general manager of the General Motors Truck & Coach Co. of Canada, Ltd., of Walkerville, Ont., Canada.

Fred W. Herlihy is now handling sales-promotion work in New York City for Shell Eastern Petroleum Products, Inc., for which he was formerly sales manager for the New England district.

Charles H. Hunt, a former engineering student at Swathmore College, Swathmore, Pa., is now employed by Starbuck & Hunt, of New York City, dealers in oriental rugs.

Albert E. Hutt is now devoting his time to his patent interests, having left his position as equipment engineer with the Vacuum Oil Co., of New York City.

Francis W. Kateley is now serving the J. G. Brill Co., of Philadelphia, in the capacity of chassis engineer in the motorcoach and motor-truck department. His prior connection was with the American Car & Foundry Motors Co., of Detroit, where he was assistant engineer. Mr. Kateley is a member of the Motorcoach and Motor-Truck Division of the Society's Standards Committee.

William H. Kelley, having relinquished his position as general superintendent with the John H. McGowan Co., of Cincinnati, is at present in Norwood, Ohio.

A. J. Langhammer, formerly factory manager for Dodge Bros. (Canada), Ltd., at Toronto, is now manufacturing and sales executive with the Amplex Mfg. Co. division of the Chrysler Corp. in Detroit.

Oscar Malychevitch, formerly chassis engineer in the engineering laboratory of the Ford Motor Co., at Dearborn, Mich., was recently appointed engineer in the Diesel division of the Hercules Motor Co., of Canton, Ohio.

A. L. Martinek, who used to be an engineer with the C. M. Hall Lamp Co., of Detroit, is now connected in the same capacity with the H. A. Douglas Mfg. Co., of Bronson, Mich., maker of electric parts for motor-vehicles.

Charles D. McCall is now connected with the sales department of the Saginaw Steering Gear division of the General Motors Corp. and is located at Birmingham, Mich. He used to be sales manager of the Muncie Products division of the corporation, with headquarters in Detroit.

Merritt A. Mieras, having resigned his position as sales engineer in the acoustical division of the Burgess Battery Co., at Detroit, is now associated with the Holley Carburetor Co., also at Detroit, in an engineering and sales capacity.

Herbert Morley, who was quality manager for the Detroit Gear & Machine Co., of Detroit, is now factory manager of the Muskegon, Mich., plant of the Norge Corp., of Muskegon Heights, Mich., a division of the Borg-Warner Corp., of Chicago.

Harry N. Parsons, formerly chief engineer of the Strom Bearing Co., of Chicago, is now chief engineer of the ball and roller-bearings division of the International Harvester Co., of the same city.

G. E. Platzer, formerly in charge of dynamometer sales for the General Electric Co., of Schenectady, N. Y., is now employed as sales manager by the Holley Carburetor Co., of Detroit.

Philip W. Remig, Jr., of Brooklyn, N. Y., has given up his position as lubrication engineer with the Vacuum Oil Co.

G. E. Ruckstell was recently elected vice-president of the Grand Canyon Air Lines, Inc., of Grand Canyon, Ariz. Prior to accepting this office he was president of the Ruckstell Corp., Ltd., of Gendale, Calif.

Frank R. Schubert concluded his contract with the Russian State bearing plant at Moscow in August and is now acting as a consultant on the design, application and manufacture of anti-friction bearings. He is located at Flossmoor, Ill.

Lester D. Seymour recently resigned his position as vice-president of the United Air Lines, of Chicago, and is now located at Sodus Point, N. Y. Mr. Seymour has given several aeronautic papers before the Society, including one entitled, Operation Experience of the National Air Transport, presented before the Metropolitan Section in 1929 and published in the S.A.E. JOURNAL for September of that year, and another entitled, Engineering and Its Relation to Aviation, presented at the May, 1932, meeting of the Chicago Section.

Henry W. Sweet has relinquished his position as special designing engineer with the Auburn Automobile Co., of Auburn, Ind., and is temporarily in Ann Arbor, Mich.

Richard R. Whittingham was appointed recently by the Standard Oil Co. of New York as sales engineer in the lubricating-oil department, working from the Boston office. He formerly served the Vacuum Oil Co. in the same capacity and district.

Glen E. Wimmer has severed his connection as engineer with the Western Electric Co., of Chicago, and is now located at Creston, Iowa.

Donald B. Wood has given up the position of transportation engineer with the Vacuum Oil Co., of New York City. Prior to accepting that position he held a similar post with General Motors South African, Ltd., in Port Elizabeth, South Africa. He is temporarily located in New Rochelle, N. Y.

Stephen J. Zand, formerly aeronautic engineer with the Pioneer Instrument Co. division of the Bendix Aviation Corp., in Brooklyn, N. Y., has been giving part of his time during the summer, and still is, to the Sperry Gyroscope Co., of Brooklyn, in the capacity of research engineer. He has given considerable study to the subject of aircraft vibration and the development of vibrographs, the latest of which he expounded in a paper presented at the Aeronautic Meeting of the Society in Cleveland last August, which was published in the November, 1932, issue of the S.A.E. JOURNAL.

Applicants for Membership

The applications for membership received between Oct. 15 and Nov. 15, 1932, are listed below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

ALLEN, FREEMAN G., district service manager, The White Co., *Chicago.*

BALDWIN, TIMOTHY, JR., sales engineer, Luce Mfg. Co., *Lansing, Mich.*

BATCHELOR, JOHN EGERTON, fuel engineer, Vacuum Oil Co., Proprietary, Ltd., *Wellington, N. Z.*

BERLINER, J. F. T., chemical engineer, E. I. duPont de Nemours & Co., Inc., *Wilmington, Del.*

BRODIE, RUSSELL W., assistant superintendent, Milwaukee Western Fuel Co., *Milwaukee.*

CARLSON, NORMAN E., owner, Ardmore Garage Repair Shop, *Chicago.*

DEATH, NORMAN P. F., mechanical adjuster, Employers' Liability Assurance Corp., Ltd., of London, *Toronto, Ont., Canada.*

EAGLES, JAMES P., draftsman, Arizone (1931), Ltd., *Sydney, Australia.*

FEATHER, M. F., sales engineer, B'ood Bros. Machine Co., *Allegan, Mich.*

GRAY, ALANSON MCDOWELL, JR., detonation engineer, Gray Processes Corp., *Newark, N. J.*

HARGRAFT, JOHN S., manager, Division C, Ontario Steel Products Co., Ltd., *Chatham, Ont., Canada.*

HILDENBRAND, KENNETH WESSELL, manager, Detroit branch, John Warren Watson Co., *Philadelphia.*

HORTON, CARROLL T., engineering assistant, Western Union Telegraph Co., *New York City.*

JONES, CERNYW E., draftsman, Perfect Circle Co., *Hagerstown, Ind.*

KIRK, BENJAMIN K., engineer, preliminary division, Baldwin Locomotive Works, *Philadelphia.*

SCOTT, A. GORDON, lubrication engineer, Imperial Oil, Ltd., *Toronto 2, Ont., Canada.*

SMITH, CHARLES A., JR., Daniels Street, *Millis, Mass.*

TEAGUE, ARNOLD L., teacher in training, San Jose State Teachers College, *San Jose, Calif.*

THURBER, LUMAN T., vice-president, The Thurberator Corp., *New York City.*

WALT, I. ROBB, teacher in training, San Jose State Teachers College, *San Jose, Calif.*

WILSON, JOSEPH M., sales engineer, Vacuum Oil Co. Inc., *Philadelphia.*

YOUNG, JOHN F., service division manager, Ford Motor Co., *Buffalo.*

ZARTMAN, JOSEPH V., technical engineer, Lincoln Oil Refining Co., *Robinson, Ill.*

News of the Sections

(*Concluded from p. 25*)

bility of engineers, particularly members of a society like our own, to exert their talents and influence in the interests of good municipal, State and Federal government. He called particular attention to the need for sane and rational consideration of taxation problems and other related matters that affect the automotive industry and various pursuits connected with the broad field of automotive transportation on our highways. He repeated the assertion that engineers should regard cooperation in public affairs of this nature as not only a privilege and a great opportunity but definitely a matter of bread and butter for themselves and their families.

In the course of Mr. Warner's remarks on the activities of the Society, the speaker paid a tribute to the life and accomplishments of Fred S. Duesenberg, who was a member of the Indiana Section for many years. He stated that in a great many ways Mr. Duesenberg typified the best characteristics of the Society, and he directed attention to the practical accomplishments of this notable engineer as exemplifying the objectives and aims of the Society, particularly the warm and human attributes that characterize the relationships within the membership.

Anti-Truck Propaganda Refuted at St. Louis Section Meeting

AT a meeting of the St. Louis Section on Oct. 27, R. G. Burr, Western editor of *Bus Transportation*, presented an extensive paper dealing with the countrywide organized propaganda to develop a public opinion favorable to planned legislation at next winter's sessions of State Legislatures proposing increased taxation and more drastic limitation of size, weight and speed of motor-trucks and motorcoaches. Following delivery of the paper, the subject was discussed by a representative of the Missouri State Highway Commission, a railroad man, truck owners and a representative of a truck manufacturing company.

Mr. Burr pointed out the powerful source of the propaganda and the financial interests involved in the competition for traffic by highway transportation and listed five fallacies which he said the proponents are "trying to get over to the public." These are that (*a*) they are interested in curbing the truck and motorcoach and do not want to increase taxes on private automobiles, (*b*) trucks and motorcoaches pay no real taxes and operate with little or no regulation, (*c*) they break and destroy the highways, (*d*) they cause innumerable accidents to other vehicles and (*e*) the spread of motorcoach and motor-truck service is causing the present financial difficulties of the railroads.

Each of these points was refuted by the speaker with quotations from authoritative, unbiased sources and by statistical facts. He quoted from association executives and class periodicals to show that the intent of the agitation is to increase the cost and decrease the effectiveness of highway transportation and to indicate that the private passenger-car is not likely to be overlooked as a competitor of common-carrier passenger traffic.

Motor-Vehicles Pay Their Way

In 1931, said Mr. Burr, all motor-vehicles paid a little more than $1,000,000,000 in various taxes, exclusive of personal property and income taxes, which represented one-tenth of all taxes paid to all political divisions in the Country; and trucks and motorcoaches are estimated to have paid a quarter of this although representing a little more than one-eighth of all the vehicles. The average tax on the common-carrier motorcoach was $811.42, and on the average truck, more than $66, although 93 per cent of all trucks are of 2-ton capacity or less. Of $1,860,000,000 of road funds provided in 1930, 46.5 per cent came from motor-vehicle and gasoline taxes. Federal aid represented less than 5 per cent of the funds, and the Government has received from taxes on the motor-vehicle industry more than it has contributed for road construction. Moreover, 78 per cent of the State highway bonds floated in the last 10 years are to be repaid by motor-vehicle taxes.

The Chief of the Bureau of Public Roads was quoted to the effect that heavy trucks and motorcoaches are fully meeting all excess costs of increased thickness of road slabs and that the highways would be built just as wide and thick if no common-carrier trucks were operated.

Accident statistics compiled by the National Safety Council show that in the three years from 1927 to 1930, private-car accidents increased 37 per cent while truck accidents decreased 14 per cent and motorcoach accidents 19 per cent.

Making Their Own Hard Bed

According to the speaker, the railroad taxes amount to $1,000,000 per day, but as against this aggregate of $365,000,000 per year, the railroads derive about $400,000,000 yearly from the haulage of about 3,000,000 carloads of motor-vehicles, parts, gasoline and road materials. He also stated that eventually the railroads will be the Country's largest operators of trucks and motorcoaches, and that whatever legislation is enacted now will "be the bed in which they will later lie."

Notes and Reviews

AIRCRAFT

Interference. By E. Ower. Published in *The Journal of the Royal Aeronautical Society*, July, 1932, p. 531. [A-1]

The author, who for more than 12 years has been on the staff of the National Physical Laboratory, discusses various experiments carried out in England and elsewhere and points out that, although many factors play a part in interference phenomena of aircraft, the evidence is consistent in indicating a few outstanding principles of general application to different branches of airplane design.

Mr. Ower gives a brief résumé of the now generally established characteristics of the flow past a simple streamline body and then considers how these characteristics are changed by attaching excrescences of certain types to the body and how the body affects the excrescences themselves. The particular case when these excrescences form the wings or parts of wings is said to be one of special interest to airplane designers and is discussed at length. Nacelle-wing interference is treated and followed by a brief discussion of matters such as the effect of wing tanks, cutaway center sections and struts. Finally, some points arising in the conduct of interference experiments in the wind-tunnel are considered.

Practical Metal-Hull Construction. Published in *Aircraft Engineering*, July, 1932, p. 165, and August, 1932, p. 197. [A-1]

The author, whose name is not disclosed, is said to have had 20 years' experience with various firms in the actual construction and erection of wooden and metal hulls for flying-boats. His article is a practical exposition of the best shop practice.

Monocoque Construction in Theory and Practice. By Richard L. Schleicher. Paper presented before a meeting of the aeronautic division of the American Society of Mechanical Engineers at Buffalo, June 6 to 8, 1932. [A-1]

Three and a half years have elapsed, the author states, since the first real attempts were made in this Country to design and build all-metal monocoque airplane structures. From early designs have gradually been evolved the three types of stressed-skin construction: the full-monocoque, the semi-monocoque and the reenforced shell. Little has been done to metalize wing structures, but the fuselage and its covering have been constructed of aluminum alloy and lately of stainless steel. Since spot welding has come into prominence along with stainless steel, a decided advantage is presented. If the cost of airplane construction is to be reduced and the way made clear for quantity production, a change must be made from the present form of monocoque construction. The author, from a long experience, presents some suggestions.

Force Measurements on a 1/40-Scale Model of the United States Airship Akron. By Hugh B. Freeman. N.A.C.A. Report No. 432, 1932; 18 pp., illustrated. Price, 5 cents. [A-1]

Lift and Drag Characteristics and Gliding Performance of an Autogiro as Determined in Flight. By John B. Wheatley. N.A.C.A. Report No. 434, 1932; 10 pp., illustrated. Price, 5 cents. [A-1]

Tests of Nacelle-Propeller Combinations in Various Positions with Reference to Wings. II—Thick Wings—Various Radial-Engine Cowlings—Tractor Propeller. By Donald H. Wood. N.A.C.A. Report No. 436, 1932; 42 pp., illustrated. Price, 10 cents. [A-1]

The Effect of Area and Aspect Ratio on the Yawing Moments of Rudders at Large Angles of Pitch on Three Fuselages. By Hugh L. Dryden and B. H. Monish. N.A.C.A. Report No. 437, 1932; 12 pp., with tables and charts. Price, 5 cents. [A-1]

Wind-Tunnel Research Comparing Lateral Control Devices Particularly at High Angles of Attack. V—Spoilers and Ailerons on Rectangular Wings. By Fred E. Weick and Joseph A. Shortal. N.A.C.A. Report No. 439, 1932; 18 pp., with tables and charts. Price, 10 cents. [A-1]

A Flight Investigation of the Spinning of the NY-1 Airplane with Varied Mass Distribution and Other Modifications, and an Analysis Based on Wind-Tunnel Tests. By Nathan F. Scudder. N.A.C.A. Report No. 441, 1932; 18 pp., illustrated. Price, 10 cents. [A-1]

Drag Tests of 4/9-Scale Model Engine Nacelles with Various Cowlings. By Ray Windler. N.A.C.A. Technical Note No. 432, October, 1932; 11 pp., 10 figs. [A-1]

The Pressure Distribution over a Standard and a Modified Navy Elliptical Wing Tip on a Biplane in Flight. By Richard V. Rhode. N.A.C.A. Technical Note No. 433, October, 1932; 29 pp., 4 figs. [A-1]

The Controls at Low Hinge-Moments. By M. Pris. Translated from *Bulletin de la Chambre Syndicale des Industries Aeronautiques*, Vol. IX, No. 6, November-December, 1931. N.A.C.A. Technical Memorandum No. 685, September, 1932; 17 pp., 32 figs. [A-1]

Methods for Facilitating the Blind Landing of Airplanes. By M. Heinrich Gloeckner. Translated from *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, June 24, 1932. N.A.C.A. Technical Memorandum No. 687, October, 1932; 23 pp., 22 figs. [A-1]

Speed and Pressure Recording in Three-Dimensional Flow. By Dr. F. Krisam. Translated from *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, July 14, 1932. N.A.C.A. Technical Memorandum No. 688, October, 1932; 13 pp., 10 figs. [A-1]

The Problem of Tire Sizes for Airplane Wheels. By Franz Michael. Translated from *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, July 14, 1932. N.A.C.A. Technical Memorandum No. 689, October, 1932; 27 pp., 27 figs. [A-1]

Transmission of Heat from a Flat Plate to a Fluid Flowing at a High Velocity. By Luigi Crocco. Translated from *L'Aerotecnica*, February, 1932. N.A.C.A. Technical Memorandum No. 690, October, 1932; 15 pp., 3 figs. [A-1]

Some Ideas on Racing Seaplanes. By Giovanni Pegna. Translated from *Rivista Aeronautica*, June, 1932. N.A.C.A. Technical Memorandum No. 691, November, 1932; 24 pp., 55 figs. [A-1]

Stabilite Automatique des Avions. By Fr. Haus. Published in *L'Aéronautique*, May, p. 139; June, p. 171; July, p. 207 and August, 1932, p. 243. [A-1]

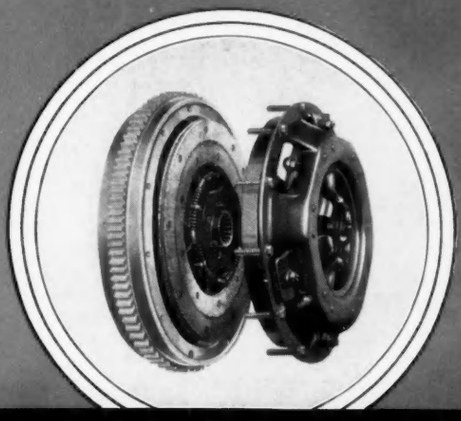
In discussing the automatic stabilization of airplanes, the author first presents the mathematical viewpoint. He shows how the disturbances to which an aircraft is subjected are translated into two systems of damped oscillations of different periods and return to the initial state of equilibrium. These two systems correspond to a movement of the center of gravity and a movement about the center of gravity. He demonstrates the impossibility of stabilizing all the variables in the present type of rigid, non-articulated aircraft structure.

Three schools of thought on the subject of attaining automatic stability of aircraft are distinguished as (*a*) those who wish to encourage the natural rotations, (*b*) those who wish to modify the direction of natural rotations and (*c*) those who wish to suppress them entirely. Descriptions are then given of current devices for attaining longitudinal stability, about 12 in number, divided into two classes, the simple, or those dealing with only one variable, and the complex, dealing with more than one variable. Their merits are

(*Continued on second left-hand page*)

These items, which are prepared by the Research Department, give brief descriptions of technical books and articles on automotive subjects. As a rule, no attempt is made to give an exhaustive review, the purpose being to indicate what of special interest to the automotive industry has been published.

The letters and numbers in brackets following the titles classify the articles into the following divisions and subdivisions: *Divisions*—A, Aircraft; B, Body; C, Chassis Parts; D, Education; E, Engines; F, Highways; G, Material; H, Miscellaneous; I, Motorboat; J, Motorcoach; K, Motor-Truck; L, Passenger Car; M, Tractor. *Subdivisions*—1, Design and Research; 2, Maintenance and Service; 3, Miscellaneous; 4, Operation; 5, Production; 6, Sales.



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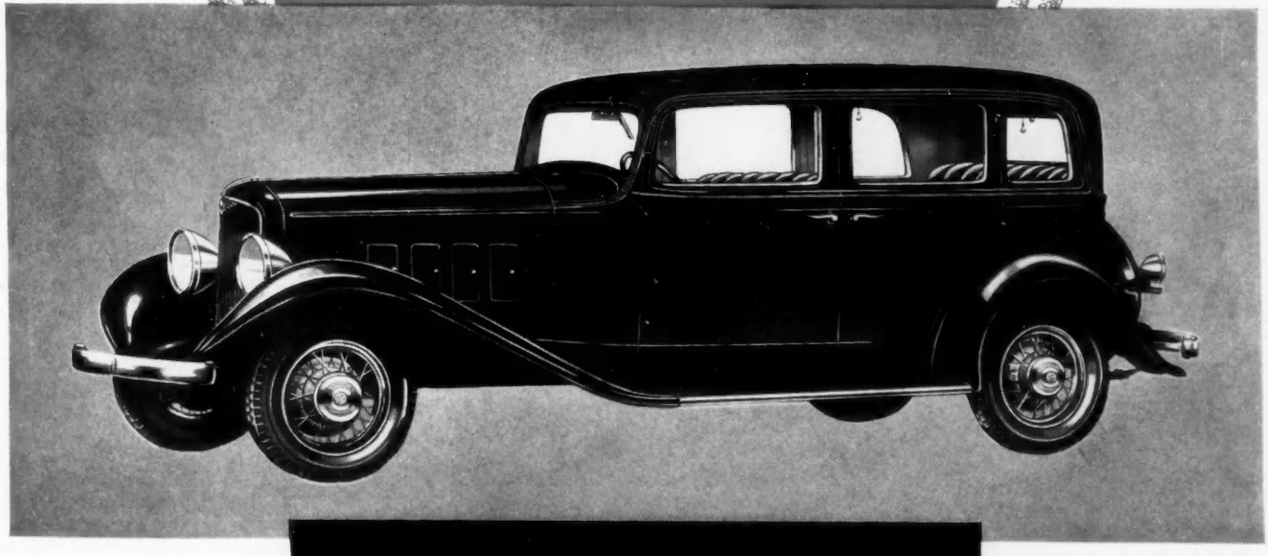

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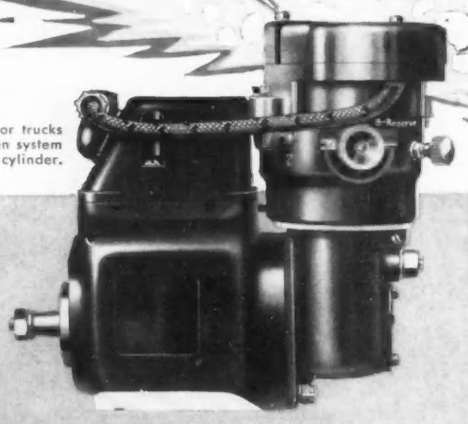

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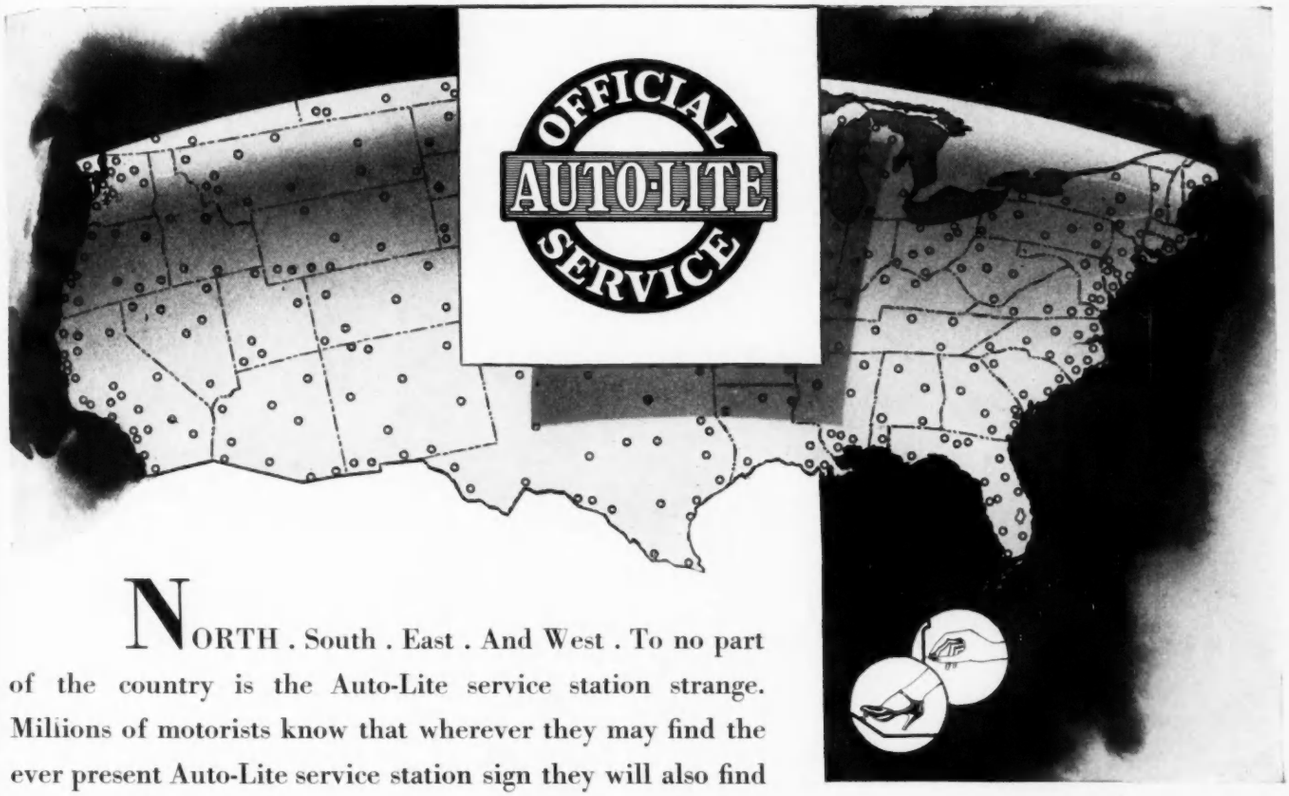

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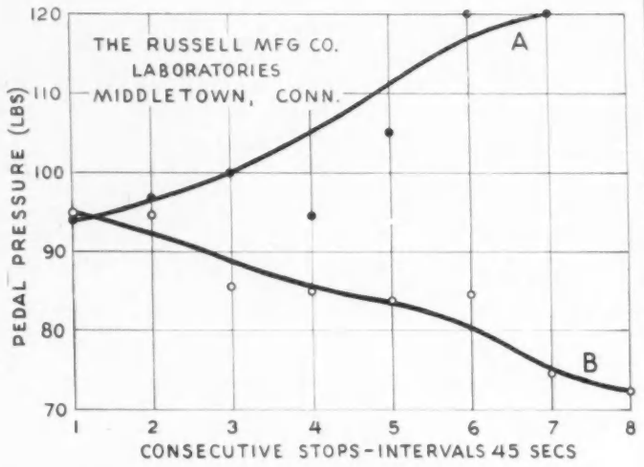

Data: Brake type, internal mechanical; drum size, 11.5 in.; drum material, steel; car speed, 45 m.p.h.; constant deceleration 15 ft. sec.²

Curve A in the accompanying chart shows a typical performance record of a Rusco Lining having *positive* fade-out, while Curve B represents the same test as made with a Rusco Lining having a *negative* fade. It will be noticed on the B curve that the pedal pressure required for the last two stops was actually *less* by about 20% than for the first two.

In these days of free wheeling it is obvious that a lining requiring less pedal pressure the longer the brake is applied offers decided advantages. Complete details will be furnished gladly upon request. Address Engineering Department B-87, The Russell Manufacturing Company, Middletown, Conn.

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BRAKE LININGS

Notes and Reviews

Continued

examined in the light of the mathematical conclusions drawn at the beginning of the treatise.

Das DVL-Gleitwinkelsteuer (Bauart W. Hübner). By Walter Hübner and Wilhelm Pleines. Published in *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, Aug. 12, 1932, p. 455. **[A-1]**

A controllable device for increasing the gliding angles of aircraft is described. It is intended especially for application to aircraft of high aerodynamic efficiency having flat gliding angles and difficult landing characteristics. This gliding-angle control, operated by the pilot, produces a local interruption in the lift distribution along the wing span and hence an increase in induced resistance.

The device consists of two rudders placed on the upper wing-surface. When not in operation they assume a position in which they do not interfere in any way with the flight characteristics of the airplane.

In flight tests the device increased the gliding angle by 60 per cent, which, for the airplane tested, signified a decrease of 40 per cent in the length of glide preparatory to landing.

Flying in the Stratosphere. By G. A. Crocco. Published in *Aircraft Engineering*, July, 1932, p. 171 and August, 1932, p. 204. **[A-4]**

This paper, presented before the 20th meeting of the Italian Association for the Advancement of Science at Milan, is described in a sub-title as a theoretical examination of the possibilities of achieving high speeds at very great altitudes.

General Crocco is well known for a long series of scientific papers on various aspects of aeronautics dating from 1903 to the present and is qualified to discuss this phase of the aviation problem.

CHASSIS PARTS

Ground-Contact Area of Tires Varies Directly with Deflection. By P. M. Heldt. Published in *Automotive Industries*, July 23, 1932, p. 100. **[C-1]**

One of the problems with which the engineers of all car-manufacturing firms are now confronted is that of low-pressure tires. This mathematical consideration of tire pressures, contact areas and load deflections is the third of a series of articles on the new low-pressure balloon tires. The previous articles, by Norman G. Shidle and James W. Cottrell, appeared in the July 9 and July 16 issues respectively of *Automotive Industries*.

Grundlagen der Federung von Automobilen. By Herbert Maruhn. Published by M. Krayn, Berlin; 111 pp., 37 figs. **[C-1]**

The non-rigid or divided type of axle construction has failed to attract the favorable attention of American and European manufacturers of large cars, in the opinion of the authors, because, with the conventional type of spring, the uniform spring action achieved with the rigid axle cannot be obtained. This fault has been regarded as sufficiently serious to outweigh the advantages of better roadability and reduced shimmy attainable through independent wheel suspension.

To combine the advantages of rigid and divided axles, the author proposes a spring layout designed by himself, which he terms "compensating suspension," for use with the latter type of construction. In principle, it consists of providing each wheel with two springs, one of which is connected directly with the frame while the other is connected to a rod that is free to rotate about the longitudinal axis of the vehicle.

In support of his thesis, the author discusses the broad fundamentals of spring suspension, with its twofold duty of shock absorption and determining wheel direction under spring deformation. He compares the relative merits of rigid and divided axles and gives examples of the design, application to passenger cars and action of compensated springs based on the principle above set forth.

This article is based on investigations made at the Berlin Engineering College.

ENGINES

Rates of Fuel Discharge as Affected by the Design of Fuel-Injection Systems for Internal-Combustion Engines. By A. G. Gelalles and E. T. Marsh. N.A.C.A. Report No. 433, 1932; 15 pp., with tables and charts. Price, 5 cents. **[E-1]**

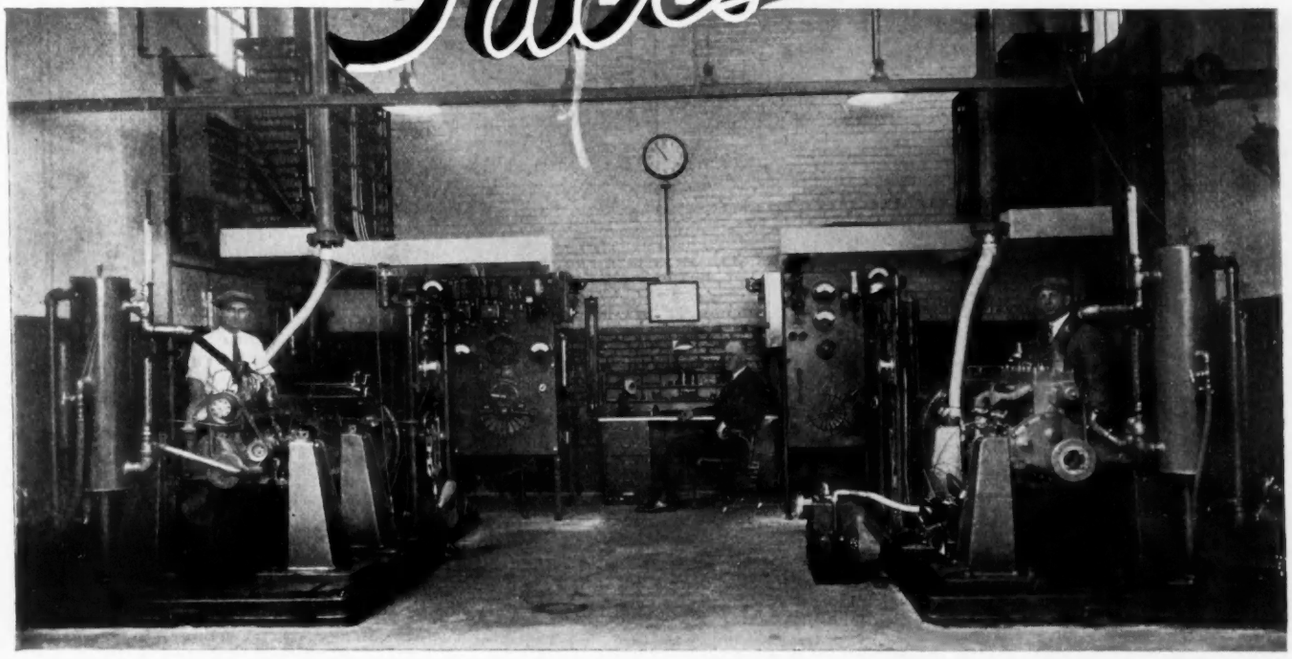
Experiments on the Distribution of Fuel in Fuel Sprays. By Dana W. Lee. N.A.C.A. Report No. 438, 1932; 17 pp., illustrated. Price, 10 cents. **[E-1]**

HIGHWAYS

Investigation of the Effect of Truck Impact on Concrete Pavements and Investigation of the Effect of Wheel Loads Applied to the Pavement by

(Continued on next left-hand page)

To The Man Who Wants The Facts



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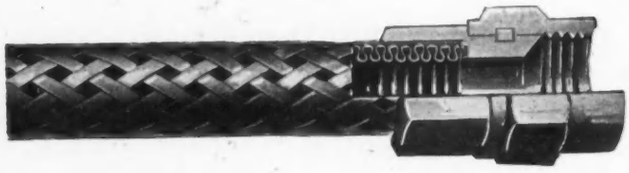

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Notes and Reviews

Continued

Six-Wheel Trucks. Reports 31-7 and 31-8 issued by the State of Illinois Department of Public Works and Buildings Division of Highways, Springfield, Ill., Dec. 31, 1931; 34 pp. and charts and 33 pp. and charts respectively. **[F-1]**

The first-mentioned investigation consists of three parts: a study of the factors that cause or influence impact, a field investigation to determine the effect of impact and the derivation of empirical formulas representing the data obtained. The formulas indicate what may be done through further investigation in the development of empirical formulas in terms of at least seven variable factors that determine the effect of impact. These factors are the sprung and unsprung parts of the load, the properties of the tire and spring, the speed of the vehicle, the radius of the wheel producing the impact and the irregularity of the surface of the pavement.

The second investigation covers the effect of loads on concrete pavements, including a study of the effect of variation in distance between two loads applied statically and a comparison of effects between four-wheel and six-wheel trucks applying static, moving and impact loads.

MATERIAL

Symposium on Spontaneous Ignition Temperatures. Spontaneous Ignition Temperatures of Fuels, by F. A. Foord; Spontaneous Ignition Temperatures of Liquid Fuels and Their Determination (German investigations), by J. L. Chaloner; Comparison of Spontaneous Ignition Temperatures and Starting and Ignition-Delay Tests in Engines, by L. J. LeMesurier and Ignition Quality of Diesel Fuels, by J. J. Broeze. Published in *Journal of the Institution of Petroleum Technologists*, July, 1932; p. 533. **[G-1]**

Mr. Foord reports the experiments made at the Royal Aircraft Establishment by W. Helmore and F. C. Code Holland. This research was started in 1928 to examine the ignition characteristics of liquid fuels, injected into air, with a view to considering chemical means of accelerating their combustion.

It was anticipated that the development of the high-speed compression-ignition engine would demand more rapid ignition of the injected fuel, and it was desired to compare the behavior of various oils in this respect to determine their relative suitability for such engines and to examine chemical means for their improvement.

Mr. Chaloner reviews the recent research work of German investigators in this field and describes the types of ignition-testing instrument in use on the Continent.

Other types of spontaneous-ignition-testing equipment and engine-starting tests are evaluated in the paper by Mr. LeMesurier.

The Cetane-number rating of Diesel fuels and the method developed by the testing laboratory of the Bataafsche Petroleum Maatschappij at Delft are outlined by Mr. Broeze, together with results obtained in attempted correlation on engines of various types.

The Effect of Jacket and Valve Temperatures on Knock Ratings of Motor Fuels. By F. H. Garner and E. M. Dodds. Published in *Engineering*, July 8, p. 45 and July 15, 1932, p. 60. **[G-1]**

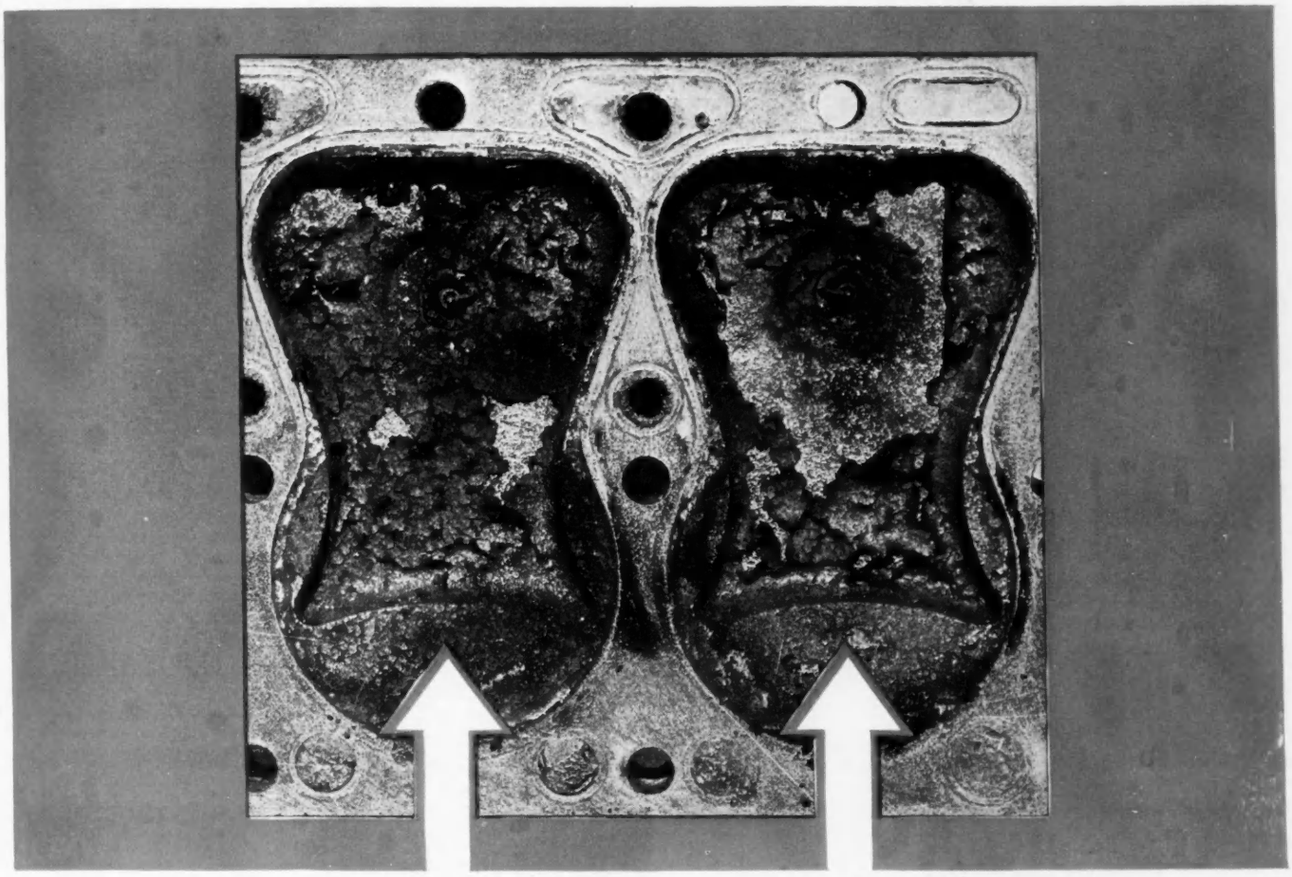
The fact is now fully realized that very careful attention must be directed to all the details of engine conditions if consistent results are to be obtained in the determination of the knock ratings of fuels. The influence of the temperatures of various parts of the engine undoubtedly is one of the most important of the several factors involved, the authors point out, adding that up to now little or no work showing the effect of varying exhaust-valve temperature on the knock ratings of fuels seems to have been recorded in print. With this particular object the work described in this article was undertaken at the engine laboratories of the Anglo-American Oil Co., Ltd., in London, on a wide range of fuels. The investigation required modification of the various engine accessories. Full descriptions are given of both the test equipment and the results obtained.

Rubber Latex. By V. N. Morris and H. W. Greenup. Published in *Industrial and Engineering Chemistry*, July, 1932, p. 755. **[G-1]**

Recent scientific and technical developments in rubber latex are reviewed. The more or less scientific work that has been done on the plantations and the problems relating to the production of articles from latex in the factories of America and Europe are discussed. A résumé is given of the actual and proposed uses for latex and products made from it. A comprehensive bibliography is appended to the paper.

(*Concluded on next left-hand page*)

GUM AND CARBON DEPOSITS



LOOSENED BY REMO

BLOW OUT WITH THE EXHAUST WHEN THE ENGINE IS STARTED

This photograph shows how REMO loosens the combustion chamber deposits which cause overheating and loss of power.

For experimental purposes, a REMO Injector was installed on an engine containing a typical accumulation of gum and carbon deposits from 2,000 miles of driving.

The engine was run until it had attained its normal operating temperature. Then a standard REMO treatment was made, just as any car owner might make it, except that the cylinder head was removed before the engine was run again, so that the action of REMO in loosening the deposits could be observed.

Notice how the combustion deposits have been loosened, in fine particles and delicate flakes, from the metal surfaces of the cylinder head. That is the way REMO acts . . . it dissolves the gummy binders which normally bind together and hold the carbon and road dust accumulations, thus loosening them so that they are free to blow out with the exhaust gases when the engine is run.

The deposits around the valves and on the pistons are loosened similarly and removed by REMO, for it spreads over all the surfaces of the combustion chambers.

REMO also cleans the gum from valve stems and prevents valve sticking from this cause. Enough of this gum-dissolving fluid also gets down around the piston rings to keep them clean and to prevent binding, or clogging of the oil-drain openings.

A REMO injection every 1,000 miles keeps an engine clean throughout its life. An AC representative will be glad to bring you full information about REMO.

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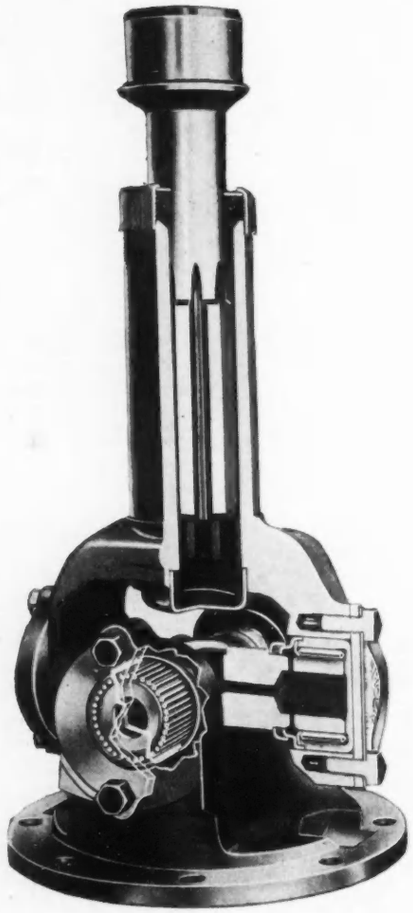

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Notes and Reviews

Concluded

Effect of Zinc Coatings on the Endurance Properties of Steel. W. H. Swanger and R. D. France. Published in the *Bureau of Standards Journal of Research*, July 1932, p. 9. **[G-1]**

The effect of the surface alterations, resulting from the application and the presence of hot-dipped galvanized and electroplated zinc coatings, on the endurance properties of 0.02-per-cent carbon open-hearth iron and 0.45 and 0.72-per-cent carbon steels was ascertained by fatigue tests made with R. R. Moore rotating-beam and Haigh axial-loading machines.

The results of the fatigue tests are given in conventional *S-N* diagrams and are summarized in a table, together with the results of tensile-strength and hardness determinations. Photomicrographs showing the structures of the heat-treated steels and of coatings of the two types are given.

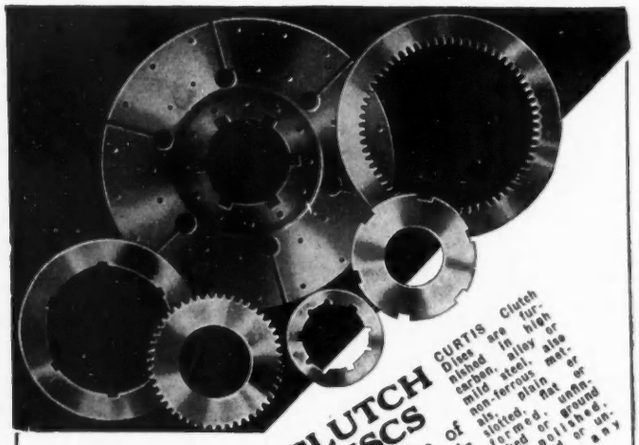
Vergütungsstähle. By Erich Eichwald. Published in *Automobiltechnische Zeitschrift*, Aug. 10, 1932, p. 363. **[G-1]**

Herein is given a summary, largely in tabular form, of special steels, their physical properties, heat-treatment and fields of use. Included are 11 steels standardized in Germany and 155 German trade-marked unstandardized steels.

La Renaissance de l'Industrie du Raffinage en France. M. Ermakoff. Published in *La Technique Moderne*, July 15, 1932, p. 458. **[G-5]**

As engineer at the national bureau of liquid fuels, the author is well fitted for his present task of making an analysis of the principles governing the laws of March, 1928, regulating the tariff on and imports of petroleum, and of their effect in promoting the renaissance of the refining industry in France. A list is given of the 11 refineries operating in France, with an indication of their total capacity. For information on the general tendencies in the refining industry, the author refers to the United States, where the cracking process overshadows other developments.

This article is the intermediate one of a series of three, the first having dealt with the chemistry of refining and the principles of refining methods, while the third will detail the equipment and processes at present used in France.



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Only long experience in the manufacture of Clutch Discs can tell a manufacturer what is necessary for satisfactory performance in automobile, truck or tractor service.

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And because of solving so many perplexing problems in this field, the Curtis Company is particularly qualified to give expert advice as to your particular needs. Correspondence is invited.

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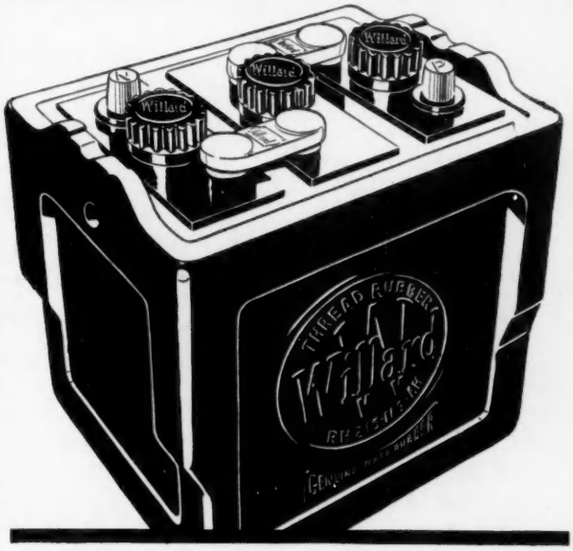

Quick Starts

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Twelve months of the most rigorous road and laboratory tests convinced Pierce-Arrow engineers that here were the brakes to give Pierce-Arrow owners the utmost in safety, dependability and driving ease.

Stewart-Warner is justly proud of this tribute to its own engineering staff.

For today—as for more than three decades—Pierce-Arrow engineering is recognized for its exacting standards and progressive conservatism. Unproved fads—even though they give promise of enjoying a tremendous sales appeal—cannot successfully run the gauntlet of Pierce-Arrow engineers.

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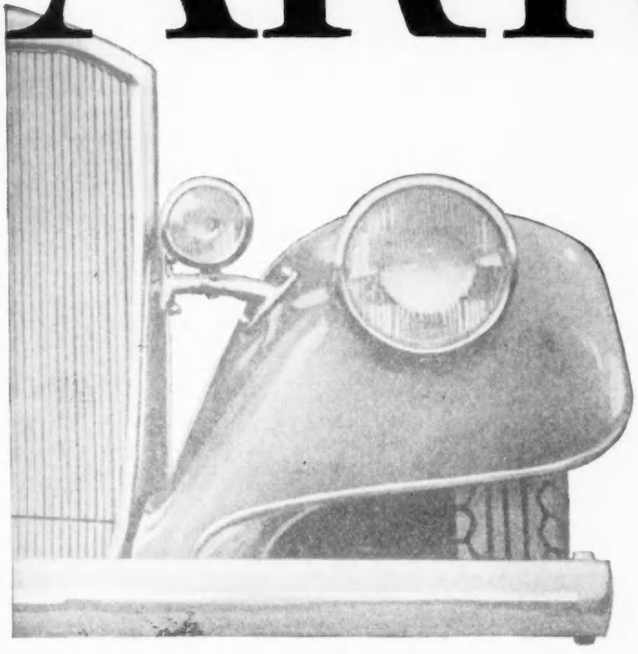
ARROW



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Before any accessory or any equipment is adopted by Pierce-Arrow, it must be proved—it must offer the ultimate to Pierce-Arrow owners.

The decision of Pierce-Arrow, therefore, is significant recognition and confirmation of the fact that Stewart-Warner Mechanical Power Brakes offer the ultimate in brake protection to meet today's driving needs.

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BRAKES WILL SELL MORE CARS IN 1933

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Advertisers whose products conform to S.A.E. specifications are also listed in the S.A.E. Handbook List of Manufacturers, beginning on page 675, of the 1931 issue of the Handbook.

The addresses of companies listed in this index can be obtained from their current advertisements indexed on page 44.

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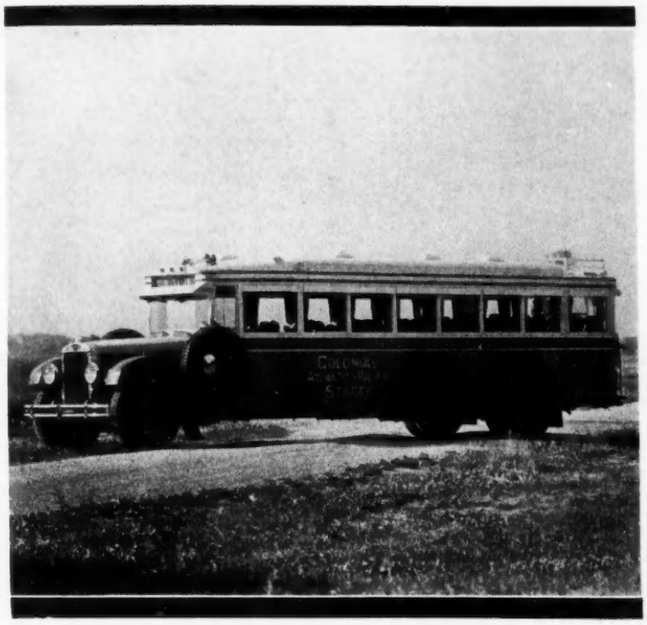

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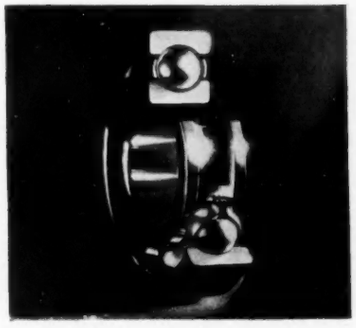

On the Mack Model BK Bus for interstate service, SKF Ball Bearings may be found in many locations from front to rear giving complete protection to equipment so necessary in long distance transportation. Throughout a long life SKF's need no adjustments, show no wear and insure silent, smooth-working units with very little maintenance. And SKF's contribute to economies because they consistently give the greatest mileage per bearing dollar.

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SKF
BALL AND ROLLER BEARINGS

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Wheels, Pressed Steel Disc
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Wheels, Wood
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ANN ARBOR, MICHIGAN

BGR INC.

DETROIT DIVISION
6400 MILLER AVENUE

BARNES-GIBSON-RAYMOND-INC

Engineering!

WHEEL ENGINEERING: Now, it must meet greater and greater demands as automotive engineering advances

TO equip the modern automobile with wheels is an assignment for only the most competent and trusted.

Increased power, higher speeds, heavier loads, more severe braking strains—all put greater responsibility on the designer and builder of wheels.

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Demountable Wood Wheels by Motor Wheel interchange with other wheel types on the same hubs

MOTOR WHEEL CORPORATION, Lansing, Michigan

Manufacturers of **Demountable Wood, Steel and Wire Wheels . . . Forged Spoksteel Truck, Bus and Trailer Wheels . . . Industrial Stampings . . .** ***Sole Producers of*** **Centrifuse Brake Drums**

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YELLOW COACHES

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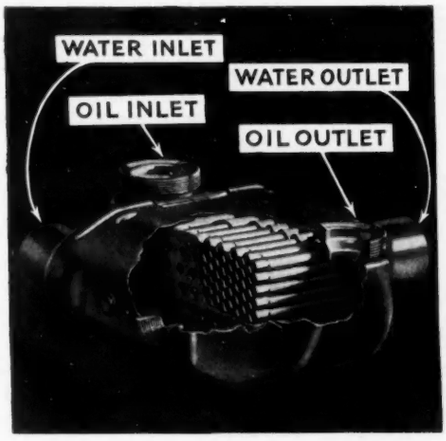

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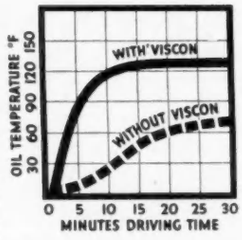

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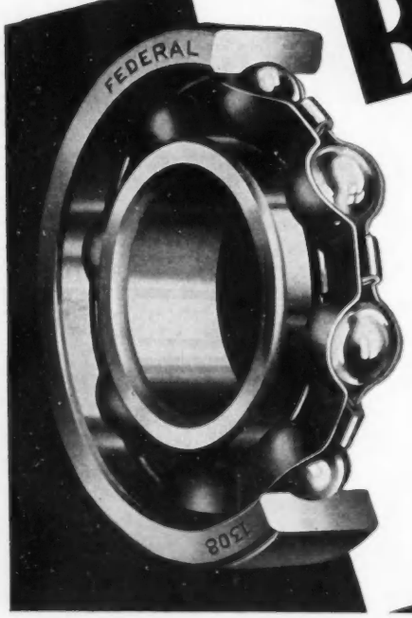

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Free-Wheeling Devices and Their Control

Semi-Annual Meeting Paper

By Austin M. Wolf[1]

DEVOTED to the mechanism of free-wheeling devices, this paper contains detailed descriptions of the devices now in use on American cars and a résumé of the experiences of the Studebaker Corp., which pioneered the device in this Country, and other prominent manufacturers, showing how present designs of roller clutch have evolved from those first developed, in which several rollers of graduated sizes were used in each pocket. Lubrication of free-wheeling transmissions is discussed.

Coil-spring clutches and one English design in which frictional contact is obtained through wedge-shaped shoes are also described.

The author concludes with the suggestion of a combination unit to include free-wheeling, service brake and sprag.

WITH free-wheeling being used on about two-thirds of the present American production chassis models, there can be no doubt that the system has merit and has been accepted by the buying public as an improvement over the conventional drive. Manufacturers of the remaining models must believe that the device has disadvantages that outweigh its advantages, but it is evident that at the moment this opinion is not held by the majority. However, the pros and cons of free-wheeling are beyond the scope of this paper, which will be devoted to the mechanism of free-wheeling devices and their control. Suffice it to say that the advantages of the device must have struck a responsive chord, otherwise it could not have reached its present popularity. Only the future can tell whether or not free-wheeling is merely a cog in the evolution of transmission design.

A free-wheeling mechanism is a device in the power-transmission line between the engine and the rear wheels which automatically allows the latter to overrun the former and automatically reestablishes the fixed driving relationship between engine and rear wheels when their relative speeds again reach their normal driving ratio. The unit can be located anywhere in the transmission line, but accepted practice has now placed it at the rear of the transmission, and it usually is a ratchet device or "mechanical valve," according to the expression coined by Constantinesco. It will be noted that this definition excludes the automatic vacuum-controlled clutch; that is merely a declutching device which, while obviating the exertion of the driver's foot on the clutch pedal, in no way affects gearshifting as does free-wheeling.

Definition of Free-Wheeling

While "free-wheeling" is a recent term, the use of overrunning-clutch devices dates back to almost the beginning of the design of machinery. Concerning vehicles, we might divide the various possible means into electrical, fluid and mechanical. The electrical method appears in most gasoline-electric drives, while the fluid type has appeared in hydraulic drives. The late C. M. Manly, once President of the Society, built a hydraulic drive with a bypass between the delivery and the return lines. A check-valve therein would correspond to a free-wheeling device. While very little work has been done on the utilization of compressed air as a medium between a compressor and an air motor, the bypassing effect can be incorporated in such a system. The atmosphere would serve as an infinitely large reservoir, obviating the need of a closed circuit. We are interested, however, only in the mechanical devices which are based on the ratchet principle. Present practice includes two types, the roller ratchet and the spring-clutch drive.

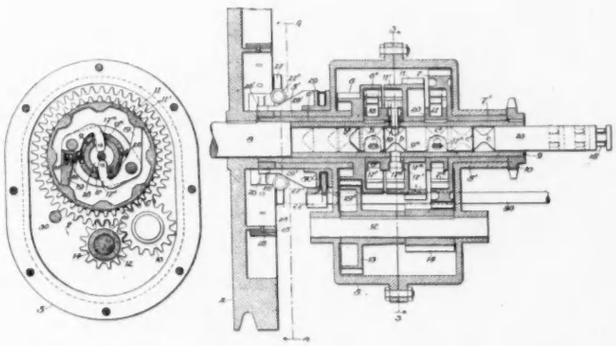


FIG. 1—PAWL AND RATCHET DRIVE OF I. H. C. BUGGY
From a Patent Issued in 1907

Free-Wheeling Devices in the Past

Information on previous developments was most ably summed up by D. G. Roos and W. S. James.[2] Comments on the situation in Europe were made by Maurice Platt at the 1928 Summer Meeting.[3] Some of the early differential substitutes incorporated pawl and ratchet mechanisms which allowed overrunning of both axle shafts, a separate locking device being used for driving in reverse. The International Harvester Co. produced a high-wheel utility vehicle, about 25 years ago, in which the actuation of a two-speed transmission depended upon the engagement of a pair of pawls for each speed, as in the construction shown in Fig. 1.

Some of the early Haynes cars, during the years 1905 to 1908, incorporated a ratchet in the countershaft constant-mesh gear. On all of the 1918 and 1919 Pierce-Arrow cars, a ratchet was incorporated in the second and third-gear cluster of the countershaft, as shown in Fig. 2. The ratchet design is of interest in view of present experience as to the rigorous ordeal to which such members are subjet.

About two years ago the Studebaker announcement of free-wheeling was followed in October of the same year by the presentation of the Roos and James paper.[2] F. S. Duesenberg had been experimenting with the L. G. S. device as far back as 1926, but the Studebaker had the field to itself in 1930. Reenforcements came in 1931, as listed in Table 1. The increased use of free-wheeling

[1] M.S.A.E.—Consulting engineer, New York City.
[2] See S.A.E. JOURNAL, January, 1931, p. 31.
[3] See S.A.E. JOURNAL, September, 1928, p. 311, and TRANSACTIONS, 1928, p. 302.

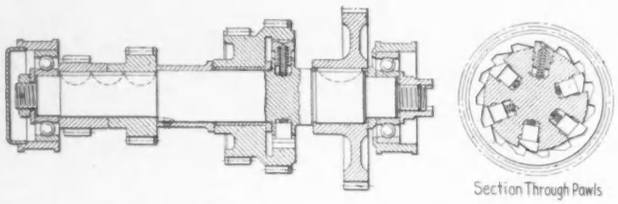


FIG. 2—RATCHET COUNTERSHAFT IN PIERCE-ARROW CARS OF 1918 AND 1919

today is strikingly borne out by Table 2, which shows that 48 models, out of 69 listed, are provided with free-wheeling.

Through the courtesy of Mr. Roos, chief engineer of the Studebaker Corp., I was given the opportunity of reviewing the company's experiences and examining experimental and production parts from the time when it pioneered the free-wheeling movement until the present. The Studebaker experiences are particularly interesting because of their influence on present-day design and production. The manufacture of a device depends on both of these interdependent items. Often devices that are easily laid out on the drawing-board present difficult problems before the completion of the finished product, and this is very much the case with the free-wheeling unit. The grief that was encountered in the earlier automotive application of roller ratchets, as in the single-unit starter and generator, discouraged many engineers, but improvements in materials, heat-treating and production methods have made the roller-ratchet unit a device of marked reliability.

TABLE 1—CARS OF 1931 PROVIDED WITH FREE-WHEELING

Make of Car	Type of Unit	Location[b]	Make of Unit
Auburn	Spring Clutch	Back	L. G. S.
Essex[a]	Roller Ratchet	Back	Warner
Graham	Roller Ratchet	Inside	Warner
Hudson[a]	Roller Ratchet	Back	Warner
Hupmobile	Roller Ratchet	Inside	Warner
Lincoln	Roller Ratchet	Inside	Own
Pierce-Arrow	Roller Ratchet	Inside	Warner
Plymouth[a]	Roller Ratchet	Back	Own
Studebaker	Roller Ratchet	Inside	Warner

[a] Announced mid-season. The other cars listed exhibited free-wheeling units at the 1931 shows.
[b] Location relative to the transmission.

The original Studebaker transmission, which is shown in Fig. 3, incorporated the free-wheeling unit in second and high gears, with a latch control in the gearshift lever to allow for going into the conventional drive after attaining the free-wheeling position. This design relieved the mechanism of low-gear and reverse-gear torque. In the opinion of S. O. White, of the Warner Gear Co., this was the most logical way to introduce free-wheeling. He states:

> The free-wheel mechanism at the rear of the transmission undoubtedly is much more elaborate than was originally contemplated; and we believe that, if we had first put the design out to the public in its present form, we would have had a very hard time getting it across. The present state has been arrived at after a course of development and education.

Experience with First Studebaker Design

Fig. 4 shows the original Studebaker cam design using three groups of four rollers. When production began, the free-wheeling unit was able to withstand 50,000 driving cycles with virtually no wear on the various parts. A car equipped with a free-wheeling transmission was operated in second gear on a chassis dynamometer by an automatic throttle-control as follows:

Starting with the car free-wheeling at a speed of 20 m.p.h. and the engine idling, the throttle was suddenly opened fully and held in this position until the car speed reached 30 m.p.h. At this time the throttle was closed and the car free-wheeled until the car speed had dropped to 20 m.p.h. again, when the cycle was repeated. The dynamometer was loaded to give resistance equivalent to that of a 5-per-cent grade.

Reports from the field showed that, after considerable mileage, slippage would occur because of grooving of the cam, especially under the two smaller rollers. To investigate this phase, a spinning test was devised in which

TABLE 2—FREE-WHEELING ON 1932 MODELS

Make of Car	Model[a]	Type of Unit	Make	Number of Rollers[a]	Model	Clutch Control
Auburn	8	Spring	L. G. S.	Double	FW 1	.....
Auburn	12	Spring	L. G. S.	Single	FW 2	.....
Buick	50	None	.....	....	.....	Regular
Buick	60	None	.....	....	.....	Regular
Buick	90	None	.....	....	.....	Regular
Cadillac	V-8	None	.....	....	.....	Regular
Cadillac	V-12	None	.....	....	.....	Regular
Cadillac	V-16	None	.....	....	.....	Regular
Chevrolet	.....	Spring	Own	Double	Chevrolet	.....
Chrysler	6	Roller	Own	10	Chrysler 6	Regular
Chrysler	8	Roller	Own	10	Chrysler 8	Regular
Chrysler	Imperial	Roller	Own	8	Chrysler Imp.	Regular
Cord	.....	.....	.....	....	.....	.....
Cunningham	V-10	Roller	Warner	8	T-82	.....
De Soto	.....	Roller	Own	10	De Soto	Extra
De Vaux	80	Roller	Warner	....	.....	.....
Dodge	6	Roller	Own	10	Dodge	Regular
Dodge	8	Roller	Own	10	Dodge	Regular
Duesenberg	.....	Spring	L. G. S.	Double	Duesenberg	.....
Essex	.....	Roller	Warner	6	AS2-FW1	Regular
Ford	4	None	.....	.....	.....	.....
Ford	8	None	.....	....	.....	.....
Franklin	S-16	Roller	Warner	8	T-81	.....
Franklin	S-17	Roller	Warner	8	T-82	.....
Graham	6	Roller	Warner	6	T-83	.....
Graham	8	Roller	Warner	8	T-81	.....
Henney	.....	Roller	Warner	8	T-82	.....
Hudson	.....	Roller	Warner	8	AS2-FW1	Regular
Hupmobile	216	Roller	Warner	8	T-81A	.....
Hupmobile	222	Roller	Warner	8	T-81A	.....
Hupmobile	226	Roller	Detroit	8	Sm-400-L	.....
La Salle	345-B	None	.....	....	.....	Regular
Lincoln	8	Roller	Own	8	Lincoln	.....
Lincoln	12	Roller	Own	8	Lincoln	.....
Marmon	125	Roller	Detroit	8	NL	.....
Marmon	16	None	.....	....	.....	.....
Nash	1060	Roller	Detroit	7	NR	.....
Nash	1070	Roller	Detroit	7	NR	.....
Nash	1080	Roller	Detroit	8	NM	.....
Nash	1090	Roller	Detroit	16	NK	.....
Oldsmobile	F32	Roller	Detroit	7	NR	.....
Oldsmobile	L32	Roller	Detroit	8	NM	.....
Packard	Light 8	None	.....	....	.....	Regular
Packard	Standard 8	None	.....	....	.....	Regular
Packard	De Luxe 8	None	.....	....	.....	Regular
Packard	Twin-Six	None	.....	....	.....	Regular
Peerless	.....	None	.....	....	.....	.....
Pierce-Arrow	54	Roller	Warner	8	T-82	.....
Pierce-Arrow	53	Roller	Warner	8	T-82	.....
Pierce-Arrow	52	Roller	Warner	8	T-82	.....
Plymouth	.....	Roller	Own	10	Plymouth	Extra
Pontiac	6	Roller	Detroit	7	NR	.....
Pontiac	B-8	Roller	Detroit	8	NM	.....
Reo	S	Roller	Detroit	8	NL	.....
Reo	8-25	None	.....	....	.....	Regular[c]
Reo	Royale	None	.....	....	.....	Regular[c]
Rockne	65	Roller	Warner	6	T-83	.....
Rockne	75	Roller	Warner	6	T-83	.....
Studebaker	6-55	Roller	Warner	8	T-81	.....
Studebaker	Dictator 8	Roller	Warner	8	T-81	.....
Studebaker	Com'd'r 8	Roller	Warner	8	T-81	.....
Studebaker	President 8	Roller	Warner	8	T-82	.....
Stutz	LAA	None	.....	....	.....	.....
Stutz	SV16	None	.....	....	.....	.....
Stutz	DV32	None	.....	....	.....	.....
Willys-Overland	6-90	Roller	Warner	6	T-83	.....
Willys-Overland	8-88	Roller	Warner	8	T-81	.....
Willys-Knight	95	Roller	Warner	10	AS2-FW4A	.....
Willys-Knight	66D	Roller	Warner	8	T-81	.....

[a] Models based on only a difference in wheelbase are not counted.
[b] "Double" and "Single" refer to the pockets in the spring-clutch type.
[c] Vacuum actuation by a Hill cylinder with foot control. Bendix automatic control is used on the other cars where clutch control is indicated as regular or extra equipment.

the shell was rotated at engine idling speed and the cam at a speed equivalent to a car speed of 60 m.p.h. The oil temperature was kept at 150 deg. fahr., allowing for a 50-deg. rise over an atmospheric temperature of 100 deg. A 1-hr. spinning test with normal oil level proved that the spinning of the rollers, and not the torque load, was responsible for the grooving. This was borne out in an extended driving-cycle test in which the cam surface was polished or burnished by the rollers over the higher portion, where the load was taken up, while grooving occurred at the low position, where the rollers would spin when free-wheeling.

The first cams were ground with a wheel travel that was parallel to the cam axis. This, of course, left ridges across the cam face. The Olsen "cloudburst" finish was tried and showed some improvement, because of the better surface obtained. Grinding circumferentially improved the finish of the cam surface, so that the wear in a 5-hr. spinning test did not exceed the wear of 1 hr. when using the old grinding method. The next step was to polish the surface after circumferential grinding.

Rollers were at first finished on a centerless grinder; then a special lapping process was added which improved the life considerably. With such a relationship of polished cam and rollers, and with suitable lubrication, 50 hr. of spinning test produced no more wear than in a 1 hr. with the original set-up.

Gears Changed To Help Lubrication

The Studebaker Corp. also found lubrication to be a most important factor. In the early development, no grooving was found after a 1-hr. spinning test with the transmission case flooded with lubricant. Various pumping schemes were tried, including vane, swash-plate and viscous pumps, in the attempt to provide force-feed lubrication for the roller-ratchet unit. These were located between the pinion shaft and the main shaft and actuated by their relative motion.

The main shaft was drilled to provide a passageway to two holes, one registering with the central position of the "pineapple" in each of its driving positions.

Various methods were tried to utilize the constant-mesh helical gears as a pump. The angles of the spiral gears were such that the lubricant was forced toward the front, and a collector ring was used between the pinion and its bearing. A centrifugal rotor was also tried, placed ahead of the bearing, to impel the lubricant to pass through the bearing to the free-wheel unit. To measure the pumping action of the constant-mesh gears with rearward flow, holes were drilled through the direct-drive clutch member that was affixed to the pinion and tests were conducted by driving the pinion in a counter-clockwise direction. This gave an abundant supply of oil to the inside of the unit, with no complications of pump, passageways or collectors. Thereupon a complete change-over in tooling was made to produce left-hand helicals instead of right-hand, and vice versa, and production was started on the revised design. A 50-hr. driving-cycle test, with peripherally ground and polished cam and lapped rollers, resulted in only a visual indication on the cam surface which could be neither felt nor measured.

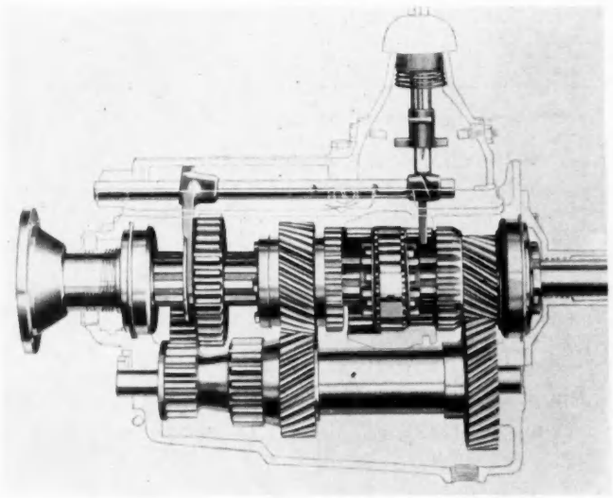


FIG. 3—ORIGINAL STUDEBAKER FREE-WHEELING TRANSMISSION

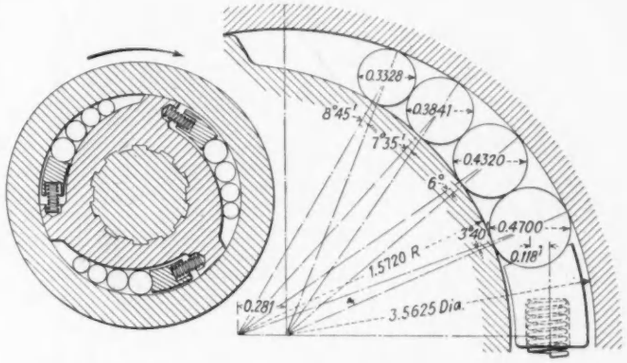


FIG. 4—CAM AND ROLLERS OF FIRST STUDEBAKER DESIGN

These developments are enumerated to show the evolution, dependent upon production methods and lubrication, in making a really successful roller ratchet. It was my pleasure to see the test parts that record each step in advance. All of this experience was valuable in the change-over to the present design and established the fundamentals that had to be adhered to. A great debt of gratitude is due to the Studebaker Corp. for allowing this review to be made.

Unit Moved to Rear of Transmission

The free-wheeling unit has now been taken out of the transmission and placed at the rear, to adapt free-wheeling to all forward speeds and provide an independent locking control instead of the previous latch incorporated in the gearshift lever with two positions for each of the two shifts. Furthermore, devices for synchronizing the gear and shaft speeds for noiseless, easy shifting were gaining ground coincident with free-wheeling developments. They are as desirable for a transmission with the free-wheeling unit as for one without, when the unit is locked out and the ease of shifting due to it is lost. For these reasons, the accepted free-wheeling transmission of today incorporates the synchronizing feature, which would have been impossible to incorporate without the relegation of the free-wheeling unit to the exterior. This seems like a reversion of type, since the original European designs had the unit in this location. However, they had entirely independent control instead of the automatic reverse-locking feature that was incorporated in American design from the start. It is interesting to note that the L.G.S. unit has always been located back of the transmission, its commercial introduction being on the 1931 Auburn Eight.

Uneven Loads on Multiple Rollers

The load carried by each roller in a roller ratchet is a function of its displacement and, since smaller rollers in the multi-roller design move most, they take most of the load. We are dealing with elastic bodies in spite of the sectional thickness of the shell and the cam. In the

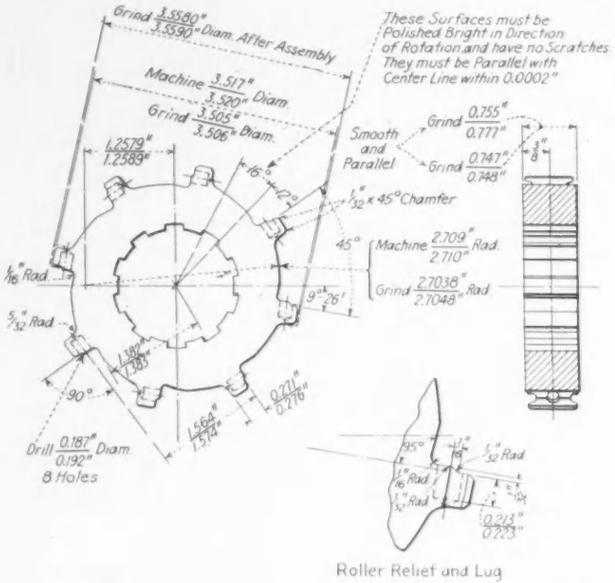

Grind 3.5580" / 3.5590" Diam. After Assembly
These Surfaces must be Polished Bright in Direction of Rotation and have no Scratches. They must be Parallel with Center Line within 0.0002"
Machine 3.517" / 3.520" Diam.
Grind 3.505" / 3.506" Diam.
Smooth and Parallel
Grind 0.755" / 0.777"
Grind 0.747" / 0.748"
3/8"
1.2579" / 1.2588"
16° 5'
1/32" x 45° Chamfer
45°
Machine 2.709" / 2.710" Rad.
Grind 2.7038" / 2.7048" Rad.
1/16" Rad.
5/32" Rad.
9° 26'
1.381" / 1.385"
90°
0.271" / 0.276"
Drill 0.187" / 0.192" Diam. 8 Holes
1.564" / 1.574"
95°
1/16" Rad.
1/32" Rad.
0.213" / 0.223"
Roller Relief and Lug

FIG. 5—CAM OF PRESENT STUDEBAKER FREE-WHEELING UNIT

Studebaker laboratory, measurements were made of the distortion in the shell with the cam held stationary and a torque load placed on the shell. The second-smallest roller showed the greatest displacement. Sometimes the shell showed a permanent set over the smallest roller. Roller diameters were varied experimentally to redistribute the load, especially to throw more load on the largest roller; but this was found to be impracticable for production, so resort was had to the single-roller design. The Lincoln, in 1931, was the first car in the field with a single roller per pocket.

The drive in the present Studebaker design is from the cam to the shell, instead of in the opposite direction as in the original design, so the present Studebaker spinning test consists in turning the cam at engine idling speed and the shell at a speed equivalent to 60 m.p.h. A cam that I examined had been through an 80-hr. test and had been only polished by the rollers, with no evidence of wear. This is a wonderful achievement when compared with the original 1-hr. test in which some wear did occur.

The present Studebaker cam, shown in Fig. 5, has its cam surfaces in the form of arcs tangent to the base circle, and the lugs which house the springs and guide the shell are integral, both as in the original cam. With a roller diameter of 0.333 in. and a shell diameter of 3.5625 in., the angle between common tangents of the roller and the cam and of the roller and the shell at the time of initial engagement is 2 deg. 30 min. The normal driving angle varies with the load; at maximum low-gear torque, the cam angle in the President transmission is approximately 5 deg. 50 min. to 6 deg. The President engine develops a maximum torque of 240 lb-ft., and the transmission low-gear ratio is 2.83:1. The unit contains eight rollers.

The lugs of the original cam had a clearance of 0.003 to 0.006 in. within the shell raceway, and locking sometimes occurred between the cam and the shell because of distortion under extreme load. Bronze caps now placed over each lug prevent any possibility of cold-welding between the cam and the shell. This seems to be a matter of small detail, but it is a decided improvement. The diametral clearance now is 0.005 to 0.007 in.

Assuming that perfect rollers are used, much depends upon their actuation. The original rollers of the four-roller design scuffed slightly unless an extreme-pressure lubricant was used. Addition of lead or sulphur in the lubricant tended to prevent this scuffing. Grinding speeds were checked to see if decarbonization of the surface could occur. Spring and shoe designs were investigated, and scuffing was overcome by a reduction in spring pressure. The spring in the original design of Fig. 4 exerted a pressure of 4 lb.; this was reduced to 1 lb. The present single-roller design uses a spring with 6 to 8-oz. pressure on the shoe. With no spring pressure, cocking of the roller due to lack of guidance caused scalloping.

Strength and Loading of Rollers

Early workers on roller clutches assumed that the state of the art in producing rollers for antifriction bearings had progressed well nigh to perfection. In the free-wheeling unit conditions are entirely different, especially when consideration is given to possible spinning speed. Taking the figures already given for shell and roller diameters, the theoretical maximum speed of a roller with the engine stalled and the car free-wheeling at 60 m.p.h. is approximately 31,000 r.p.m. Exceptional treatment is required to produce a roller that will stand up in this service. Some phases of this work will be mentioned in connection with metallurgy.

In the original free-wheeling design, the load was not equally divided among the rollers. This is largely overcome in present practice in which the rollers are all of the same diameter and operate in individual pockets. Naturally, the individual loading is reduced proportionately when a greater number of rollers is used. The load each roller must withstand is

$$L = (TR/rn) \cot a \qquad (1)$$

where

a = angle between tangents of the roller at its points of contact with the cam and the roller in degrees
L = load per roller in pounds
n = number of rollers
R = transmission reduction ratio
r = radius of circle passing through centers of rollers in inches
T = maximum engine torque in pound-inches

Assuming a cam-engagement angle of 6 deg., the load for the Studebaker President model is 5942 lb. per roller, according to Formula (1). As the roller is 3/4 in. long the load per inch is 7923 lb. While friction in the transmission and elasticity in the various elements

TABLE 3—MAXIMUM TORQUE CAPACITIES OF FREE-WHEELING UNITS

Make of Unit	Model	Torque Capacity, Lb-Ft.[a]	Remarks
Warner	T-83	450	6 Rollers
Warner	T-81	600	8 Rollers
Warner	T-82	795	8 Rollers
Detroit	NR	450	7 Rollers
Detroit	NM	555	8 Rollers
Detroit	NK	675	16 Rollers
Detroit	NL	675	8 Rollers
L. G. S.	FW1	499	Double Pocket
L. G. S.	FW2	790	Single Pocket
L. G. S.	Duesenberg	1,331	Double Pocket

[a] Capacities of the roller clutches are not very exactly defined by their manufacturers, who rate them conservatively. The figures given presuppose a low-gear reduction of 3:1. Ratings of the spring clutches are based on their ultimate capacity (the point at which the elastic limit is reached and the most heavily loaded turn of the spring is slightly distorted, causing additional frictional drag in the clutch when overrun), allowing a factor of safety of 4.

will decrease the computed loading, shock loads will augment the figures considerably, and Formula (1) should be used for normal conditions. According to figures supplied by C. F. Rauen, of the Detroit Gear & Machine Co., pressures of 400,000 to 600,000 lb. per sq. in. exist at the line of contact between the rollers and the cam and shell as the unit assumes the driving position.

Recommendations Based on Experience

Through the courtesy of Erik Oberg, editor of *Machinery*, I was able to obtain some advance information on an article that soon will appear in that publication and will bear principally on the manufacture of rollers as practised by the Bantam Ball Bearing Co., of South Bend, Ind. K. L. Herrmann, of that company, has done some extraordinary work in the development of rollers for free-wheeling units, and the article mentioned will be a valuable contribution to the art. Among the rollers manufactured at the Bantam plant are the 0.333 x 0.740-in. Studebaker rollers. For a static test, these rollers are tested by the three-in-a-row method and will support 47,000 lb. without breaking. This amounts to 62,666 lb. per in. of roller length. For a drop-hammer rating, a 20-lb. weight is dropped from varying heights, the roller being held immovable in a fixture. The height is sufficient to give an impact of 100 ft-lb. for the first drop, 110 ft-lb. for the second, 120 ft-lb. for the third and 130 ft-lb. for the fourth. The specification requires that this roller must resist the third drop at 120 ft-lb.

As a result of an immense amount of experimental work, S. O. White, of the Warner Gear Co., has arrived at 25,000 lb. per in. of length as the maximum safe roller load for operating conditions. The figure 7923 lb. per in., evolved from Formula (1), is a little less than one-third of the safe figure given by Mr. White and about one-eighth of the ultimate strength shown in the Bantam static test. Mr. White states:

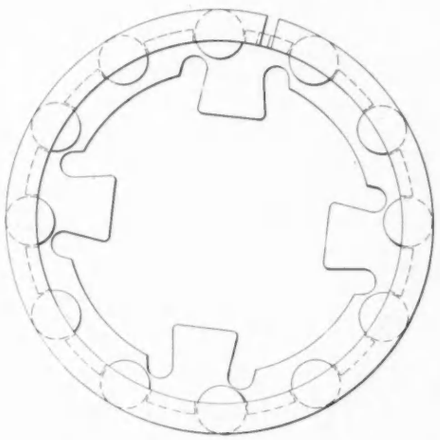
> The exact diameter of the roller, within a certain range, makes no particular difference. Some thought that the smallest roller in last year's design—in which we had multiple rollers, the diameters of which were 0.3333, 0.384, 0.432 and 0.470 in.—probably would indent the cam surface more than the largest roller. We therefore ran a large number of tests in a test machine, using an actual load of 15,000 lb. on each roller, which corresponds to about 22,000 lb. per in. of roller length. The indentation into the cam surfaces of C62 hardness over a range of tests was the same, within 0.0001 in., for rollers of all sizes. We concluded that a smaller roller, having less mass and greater elasticity, would be distorted under load more than the larger roller.

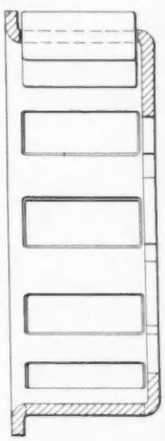
The Warner Gear Co. makes its roller shoes of zinc-base die-castings. They cover slightly less than one-quarter of the circumference of the roller. Mr. White believes this to be a matter of convenience largely and that they undoubtedly could cover less surface and still be satisfactory. There seems to be no tendency for them to chatter.

Development of Chrysler Design

I am indebted to H. T. Woolson for the information regarding the free-wheeling activities of the Chrysler Corp. Its first design embodied a three-lobe cam having three rollers per lobe, graduated in size as in the original Studebaker design. Instead of integral lugs, the Chrysler cam had screwed-on abutment blocks or lugs. Each bank of rollers was engaged by a spring. Mr. Woolson states:

> The difficulty experienced with this type was to get all rolls functioning as intended. Because of the cam design, the small rolls engaged first and carried most of the load. Shell and cam have been indented under the small rolls. Contact angle was about 3½ deg. for the large roll, 6 deg. for the medium roll and 8 deg. for the small roll. The rolls also made contact between each other, so that one spring could be used to energize the group of three. The small roll sometimes slipped on the cam, because of the high angle, the rest of the rolls slipping too, because the mutual contact did not allow rollers to be engaged independently. In addition to indenting and slipping, there was wear from roll spin, which was emphasized by heavy springs back of each group of rolls. In manufacturing, it was almost impossible to keep the diameter of rolls such that the engagement would be simultaneous.
>
> After using three-lobe cams with three rolls, we reduced the number of rolls per lobe to two, which was more satisfactory for manufacturing. After using two rolls per lobe for some time in production, we concluded that the most satisfactory job would be one roll per lobe, and from that point on our experience has been interesting. It is significant that the earliest type of three-lobe cam carried four different-sized rolls; and, as field experiences increased, the number was reduced to three, then two and eventually one. Apparently, we all started at the wrong end. Another objection to the three-lobe cam was that sometimes one bank of rolls would come in ahead of the others and the reaction load was then thrown on the pilot bearing, causing indenting and other troubles. When the roller-retainer type was adopted, this difficulty was eliminated.





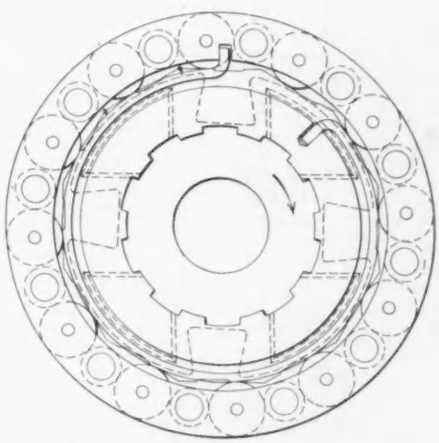


FIG. 6—ROLLER CAGE NOW USED BY CHRYSLER. FIG. 7—CHRYSLER EXPERIMENTAL CAM AND ROLLER ASSEMBLY

The Cage Shown in Fig. 6 Has Replaced in Production the Cage Shown in Fig. 7

After we began to use only one roll per lobe, we found that the possible number of rolls on a given size of cam was restricted by the abutments on the cam, and we therefore began research work on the cam-roll retainer, eliminating the cam abutments. This type is now in production. The development went through many more phases than we originally anticipated, but the result of the work was well worth the time and money spent, as we secured a cheaper, lighter and more efficient unit having more, longer and smaller rolls and a very cheap cam. The stamped retainer shown in Fig. 6 makes all rolls do their work and keeps them squarely across the cam. It also works well with a single, light energizing spring attached to the roller cage and the cam.

Fig. 7 is a diagram of the assembly with an experimental roller retainer consisting of two rings spaced by round bars between the rollers. Mr. Woolson continues:

FIG. 8—CHRYSLER FOUR-SPEED TRANSMISSION, WITH FREE-WHEELING UNIT

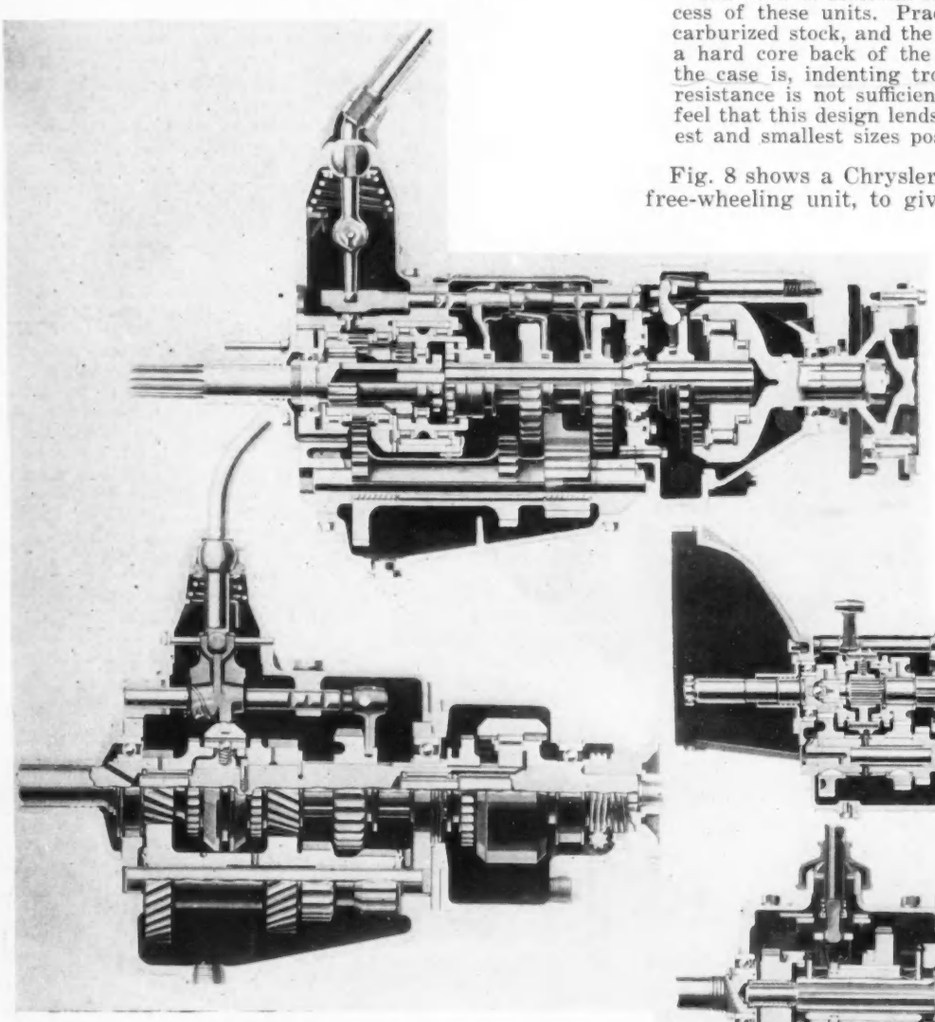


FIG. 9—WARNER T-81 TRANSMISSION, WITH FREE-WHEELING

To have the free-wheeling of this design work only in one direction, the rear side of the roller retainer is provided with four lugs which extend into cam slots. As soon as the cam slows down in respect to the driven shell, the roller retainer as a unit is moved back on the cam until the lug strikes the side of the cam slot and stops the backward movement of the rolls in respect to the cam, thus preventing reverse locking of the free-wheeling unit. Each cam lobe has a relief to give the roll sufficient clearance in free-wheeling position; otherwise the rolls would spin and the free-wheeling unit would be noisy.

Our small cams have as many as 13 rolls up to 1 in. long. The rolls could be made longer if desired, and as small as ¼ in. in diameter. It is obvious, assuming that all rolls do their work as they do in the retainer type, that the more and the longer the rolls are, the lower are the unit pressures and the smaller can be the sizes of the other parts.

The size of our units is now controlled by the cam size and the thickness of material required between the roll and the spline. We have been able to reduce the sizes materially and have a much higher factor of safety than we had in the larger units for the same service. We have found absolutely no indenting, and cam wear from spinning of the roll is entirely eliminated because of the very light energizing spring required on the cage.

The kind of material has much to do with the success of these units. Practically everyone is using a carburized stock, and the main consideration is to get a hard core back of the case. No matter how hard the case is, indenting troubles will occur if the core resistance is not sufficient. All things considered, we feel that this design lends itself to the lightest, cheapest and smallest sizes possible in roller-type clutches.

Fig. 8 shows a Chrysler four-speed transmission and free-wheeling unit, to give an idea of the general as-

FIG. 10—HUDSON TRANSMISSION UNIT

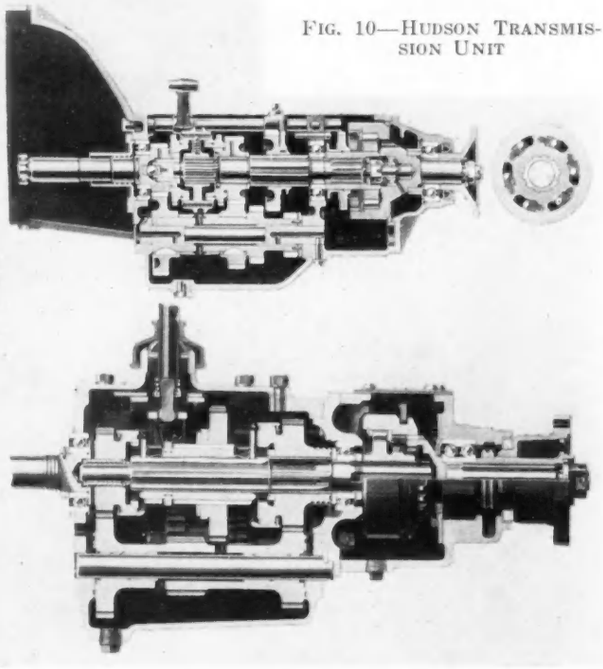


FIG. 11—OLDSMOBILE TRANSMISSION, WITH WARNER FREE-WHEELING UNIT

sembly. The stamped retainer is not shown, because of its recent release for production. The angle of initial engagement of the roller is 3½ deg.

Warner and Detroit Products Similar

The Warner transmission with free-wheeling unit is shown in Fig. 9. A rectangular key at the rear of the main shaft passes under the rear transmission bearing, and the rear end of the key contacts with the face of the free-wheeling locking-gear. The reverse sliding-gear contacts with the forward end of the key after it has gone about one-third of its way into mesh with the reverse idler. Further movement pushes the locking gear into mesh with the internal clutch teeth of the free-wheel shell, thus locking the free-wheel unit. A spring is compressed at the same time to return the locking gear

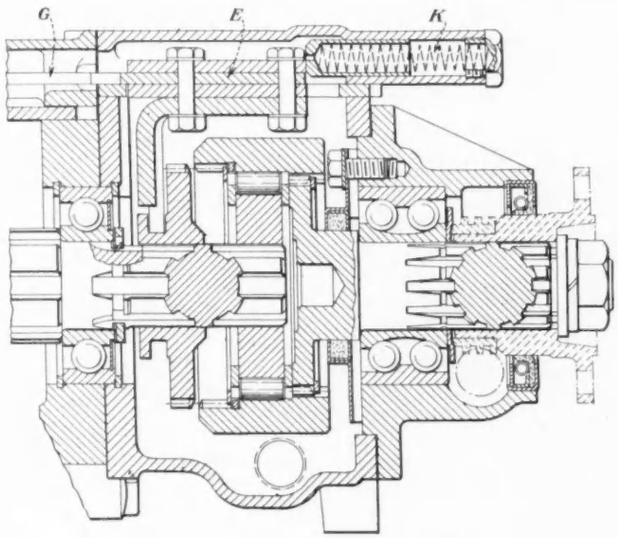


FIG. 12—DETROIT SM-400-L ASSEMBLY

Reference Letters Are Used in the Description of the Control Mechanism. See also Fig. 21

to the position shown in Fig. 9, when reverse is disengaged. Warner transmissions are used on a number of cars, as indicated in Table 2. The Warner company also supplies the free-wheeling unit only to the Hudson and Willys-Overland companies. In the Hudson construction, shown in Fig. 10, the transmission case is extended to inclose the free-wheeling unit, instead of the usual construction in which a separate case is bolted to the rear of the transmission. Fig. 10 also shows that the low-reverse gearshifting fork, when moved backward, contacts with a plunger, the movement of which meshes the locking gear with the internal-gear portion of the free-wheeling shell.

A number of automobile manufacturers are using the Detroit Gear & Machine Co.'s unit, as will be seen from Table 2. The cam angles are the same as those used by the Warner Gear Co., as the latter company has conducted extended researches and the companies are affiliated through the Borg-Warner Corp. Variations in Detroit designs are that the NR unit has 7 rollers mounted on 7 lobes, the NM unit has 8 rollers on 8 lobes and the NK unit has 16 rollers mounted on 8 lobes, with 2 rollers side by side on each lobe. In this unit are also 16 shoes and 16 springs. Fig. 11 shows the Oldsmobile transmission with the Detroit free-wheeling unit at the rear. In this the shell drives the universal-joint shaft by an external-internal-gear fit. The shell is also movable and the cam travels with it. To lock out the free-

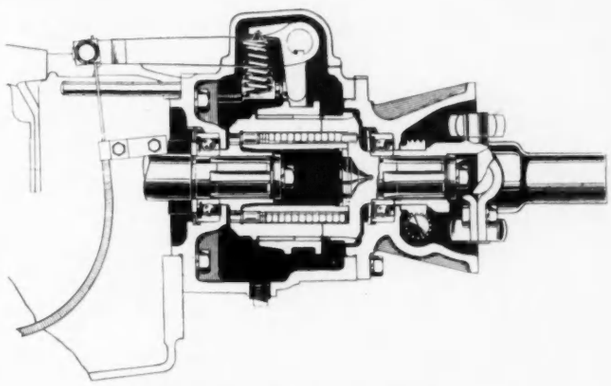


FIG. 13—CHEVROLET FREE-WHEELING UNIT

wheeling action, the whole unit is shifted to the rear and an external gear on the cam member engages an internal gear at the forward end of the universal-joint shaft.

The Detroit Gear & Machine Co. also manufactures the SM-400-L transmission, shown in Fig. 12, which incorporates a fixed free-wheeling unit and a sliding-gear clutch for locking. With this construction, which is being used in the later units, less inertia must be overcome in shifting and a smaller shift fork can be used. The reverse-locking action is accomplished by an ingenious mechanism which will be described in connection with controls.

Spring-Clutch Type

The old Mercedes band-type clutch, in which the band contracted on a flywheel drum, will be recalled by many. Its method of operation is described in textbooks as belt or coil friction. The same principle is equally applicable to a contracting or wrapping coil or an expanding one. This construction is utilized by the L.G.S. Devices Corp. for a free-wheeling device. Fig. 13 is a section of the Chevrolet spring-clutch assembly, of this type, which shows its double-pocket construction. Fig. 14 shows the three essential parts that make up this device. A similar construction is used in the Auburn Eight. The single-pocket type, shown in Fig. 15, is a new development which is used on the Auburn Twelve. The single-pocket design is shorter, having less turns in the spring, is smaller in diameter and has a smaller locking gear. Furthermore, the most heavily loaded coil is located at the blind end of the pocket, instead of crossing the open ends of the pockets as in the case of the two-pocket design.

When the device is overrunning, the main coils of the

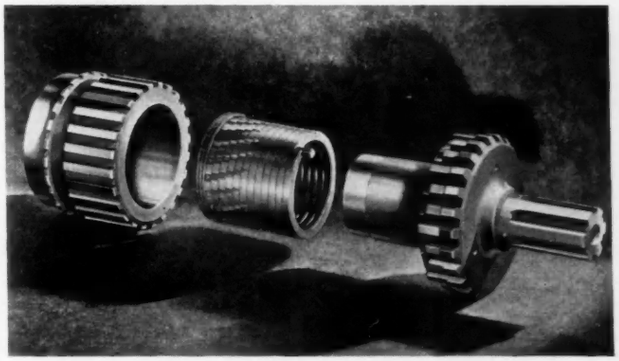


FIG. 14—MAJOR PARTS OF CHEVROLET DEVICE

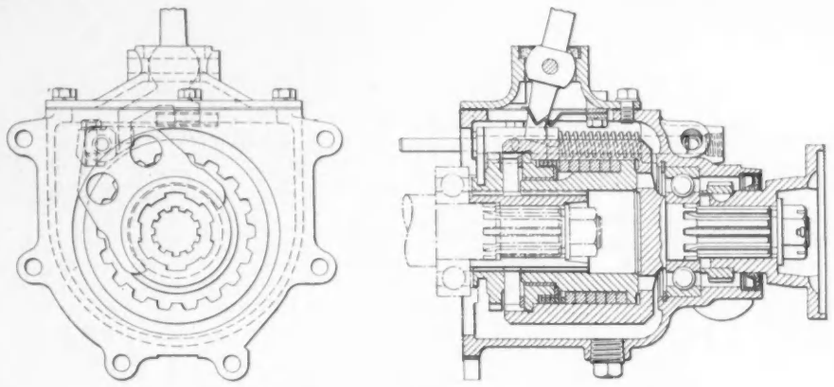


FIG. 15—FREE-WHEELING UNIT OF AUBURN TWELVE

clutch spring are under tension on the inner driving sleeve and clear the revolving outer cup by 0.004 in. When the clutch engages, the spring expands and is 0.004 in. loose on the inner sleeve. One object of the inner sleeve is to limit the contraction of the spring and thereby reduce the backlash. An energizing portion of the spring at the forward end is made of smaller wire. It is always in contact with the outer cup and some heat is developed at this point when free-wheeling at high speeds, but not enough to interfere with lubrication. When the main coils expand against the outer cup to take up the drive, part of the oil between these parts is squeezed out, the spring being provided with slots for this purpose. The drive is through a lug on the inner sleeve and then through the spring and by friction to the outer cup. This lug bears against the end of the wire. In the Chevrolet, Auburn-Eight and Duesenberg designs, the last being of the double-pocket type, the drive is through the outer sleeve, then to the spring by frictional contact and then to the shallow pocket or driven member by the spring contact and the offset in the end of the spring, which fits into a drilled hole.

Springs of Various Designs

The Duesenberg design, as shown in Fig. 16, has a special type of spring in which the wire section is gradually reduced from the central portion to the free end. This type more nearly approaches the theoretically correct spring, and it is very expensive to manufacture. The spring is straight and is forced into a pocket tapered slightly at the free end of the spring to provide sufficient friction to energize the spring. This eliminates the smaller or servo spring of the other constructions. The Duesenberg spring is anchored by being bent and fitted into a hole. It is figured that, under usual conditions, only about 5 per cent of the torque load is applied to the bent end. Another difference in design is that the locking device is at the rear end of the unit.

The regular production L.G.S. spring, while originally of uniform section, is made to approach the Duesenberg tapered type by grinding slots of varying depth, as shown in Fig. 17. This gives equally good results and is much cheaper to make, although it is slightly longer for a given number of turns.

The capacity of the spring clutch can be computed by the formula

$$P = p \times e^{f\theta} \qquad (2)$$

in which

$e = 2.718$ = base of Naperian logarithms
f = coefficient of friction
P = total torque capacity of the spring clutch in pound-inches
p = torque applied to the free end of the spring in pound-inches
θ = angular length of the spring in radians (1 turn = 6.2832 radians)

It also follows that

$$p = P/e^{f\theta} \qquad (3)$$

Table 4, showing the value of $e^{f\theta}$ for various conditions, is given through the kindness of the L.G.S. Devices Corp., which made experiments indicating that the average coefficient of friction is about 0.20. The capacity of any of its clutches is approximately proportional to the cube of the diameter of the spring.

Automatic reverse locking of the L.G.S. units is obtained by the rearward movement of the transmission low-reverse shifting rod, which strikes the shifting fork of the unit when the reverse gear is partly meshed. A spring surrounding the fork rod is compressed to unlock the unit subsequently.

Metallurgy of the Parts

To prevent indentation of the rollers or wear in the cam, the Warner Gear Co. makes both the cam member and the shell of S.A.E. 4615, with very careful attention to heat-treatment. They provide a hard case with a dense body immediately below it. A minimum hardness of C61 Rockwell is required on these surfaces. The rollers are kept slightly softer, to prevent crushing or flaking, the range being C56 to C60 Rockwell. According to the forthcoming article of Mr. Oberg, before mentioned, S.A.E. — 52100 chromium steel coming from the mill in round bars is cut off to length and each end is chamfered by a coining process before the first tempering. This is followed in order by centerless grinding of the diameter, end grinding, the second tempering, finish grinding and lapping very carefully to remove the roughness produced in the centerless grinder. The rolls are held to a tolerance of ± 0.0001 in.

The usual practice in making rollers is to quench them from a temperature 20 to 30 deg. higher than in the process developed by the Bantam Ball Bearing Co., and with this temperature the hardness of the rolls is approximately C66 Rockwell. In the usual process they are then drawn back to the hardness required, which

TABLE 4—VALUES OF $e^{f\theta}$ WITH VARIOUS COEFFICIENTS OF FRICTION, FOR USE IN EQUATIONS (2) AND (3)

No. of Turns of Wire	Turns in Radians	Coefficient of Friction 0.10	0.15	0.20	0.25	0.30	0.35	0.40
1½	9.425	2.56	4.11	6.59	10.6	16.9	27.1	43.4
1¾	10.996	3.00	5.20	9.02	15.6	27.1	46.9	81.3
2	12.566	3.50	6.56	12.3	23.0	43.0	80.5	151
2¼	14.137	4.11	8.34	16.9	34.3	69.5	141	286
2½	15.708	4.81	10.6	23.1	50.8	111	244	536
2¾	17.279	5.63	13.4	31.7	75.1	178	423	1,004
3	18.850	6.59	16.9	43.4	111	286	733	1,881
3¼	20.420	7.71	21.4	59.4	165	458	1,270	3,526
3½	21.991	9.02	27.1	81.3	244	733	2,201	6,609
3¾	23.562	10.6	34.3	111	362	1,174	3,814	12,388
4	25.133	12.3	43.3	152	536	1,881	6,609	23,221
4¼	26.704	14.4	54.9	209	793	3,031	11,453	43,526
4½	28.274	16.9	76.7	286	1,174	4,827	19,846	81,586
4¾	29.845	19.8	87.9	391	1,739	7,733	34,389	152,930
5	31.416	23.1	111	536	2,575	12,388	59,591	286,650
5¼	32.987	27.1	141	733	3,814	19,846	103,260	537,310
5½	34.558	31.7	178	1,004	5,648	31,716	178,938	1,007,138
6	37.699	43.4	286	1,881	12,388	81,568	537,310	
6½	40.841	59.9	458	3,526	27,171	209,372	1,613,349	
7	43.982	81.3	733	6,609	59,591	537,310		
8	50.266	152	1,881	23,221	286,650			
9	56.549	286	4,827	81,586	1,387,867			
10	62.832	535	12,388	286,650				

may range from C56 to C60, the average total amount of drawback being about 8 points. In the Bantam process, the original quenching hardness ranges from C60 to C62. The rollers are then drawn back to C57 to C59, making the total drawback hardness-change 3 points on the Rockwell scale. Mr. Herrmann finds that this is a very important point, as rolls having surfaces of the hardness thus obtained, rather than dead hard from the quench, are elastic.

Mr. Herrmann insists upon having the rods spherodized in the steel mill after they are rolled. Spherodizing is a type of annealing or heat-treating consisting of normalizing the steel at the usual temperature for 48 hr. and cooling it in a specific series of steps. This process results in uniformity for machining and uniform response to heat-treatment in the final quenching and drawing operations, all of which tends to produce uniform strength and eliminate the possibility of any of the rollers being cracked or strained. This process leaves the cementite in globular form. Because the rollers are never dead hard, they are elastic as they cool in the quench and do not have the surface strain which goes progressively from the surface toward the center when the critical or shrinkage range is passed.

The clutch members of the L.G.S. units are made from case-hardened alloy steel, usually S.A.E. 4615. The single pocket and driving sleeve are forgings, and the locking gear and adapter are machined from bar stock. Oil-treated alloy steel also can be used for the pocket and sleeve, and their hardness should be at least 10 Rockwell points harder than that of the spring. The Chevrolet overrunning sleeve is held to C58 to C62. The shallow pocket is slightly softer, the minimum being C49. The spring is made of tempered S.A.E.-6150 chrome-vanadium wire and drawn to a hardness of C43 to C48 Rockwell. The operations in the manufacture of the spring are about as follows, according to D. T. Brownlee, of the L.G.S. Devices Corp.: wind, normalize to relieve strains, grind longitudinal slots, grind outside diameter, drill hole in end for insertion of the energizing spring, and secure the energizing or servo spring.

Lubrication of Roller-Clutch Units

Almost any lubricant within reason will do when a roller ratchet is new and engagement is perfect. When wear in the cam surface results from spinning, the local cam angle is altered. The viscosity effect of the lubricant tends to prevent the roller from passing, in engagement, beyond the depression thus formed. If the resistance is great enough to prevent the roller from passing the depression, engagement is prevented and slipping occurs. Therefore the prime essential in a lubricant is to guard against too great viscosity, in order to prevent slippage. A light, winter engine-oil would fill this requirement for the roller ratchet, but gear rattle and noise is excessive if the same oil is used for the transmission. The selection of a lubricant therefore involves a compromise in viscosity between that required to prevent slipping and that required to prevent gear noise at high operating temperature.

Some manufacturers are of the opinion that the use of an extreme-pressure lubricant will postpone wear on the cam surface, but there is still argument as to whether certain types of extreme-pressure lubricant really will reduce the wear between roller and cam or whether the straight mineral product should be used.

The oil with the flattest viscosity curve is favored, in view of temperature changes that must be encountered and the direct relationship between temperature and viscosity. Some of the oil people have found that naphthene-base oils allow less slippage than paraffin-base oils, and oils containing soap stock allow still greater slippage.

When the roller engages, the pressure caused by the

FIG. 17—L.G.S. SPRING FOR USE IN A SINGLE POCKET

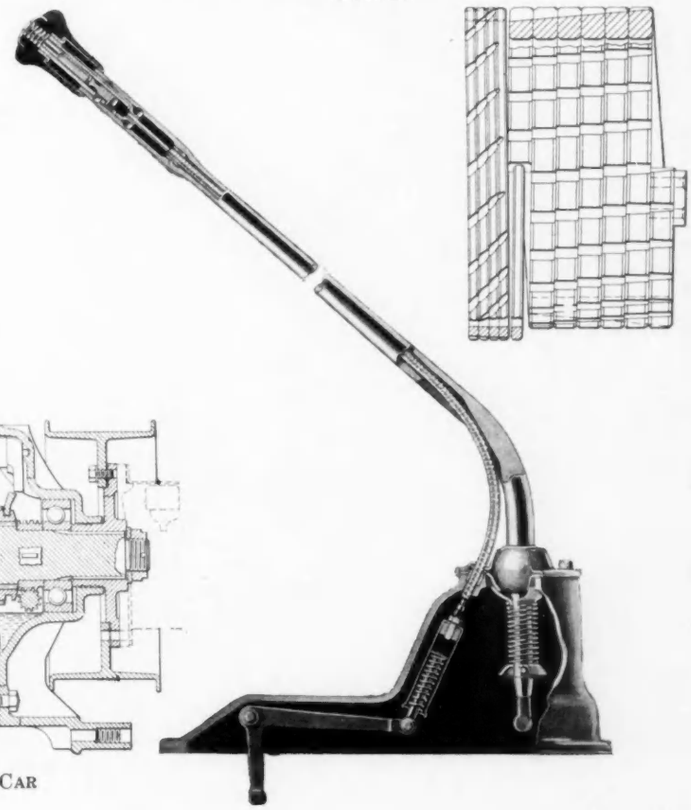


FIG. 18—HUDSON GEARSHIFT LEVER, WITH FREE-WHEELING CONTROL

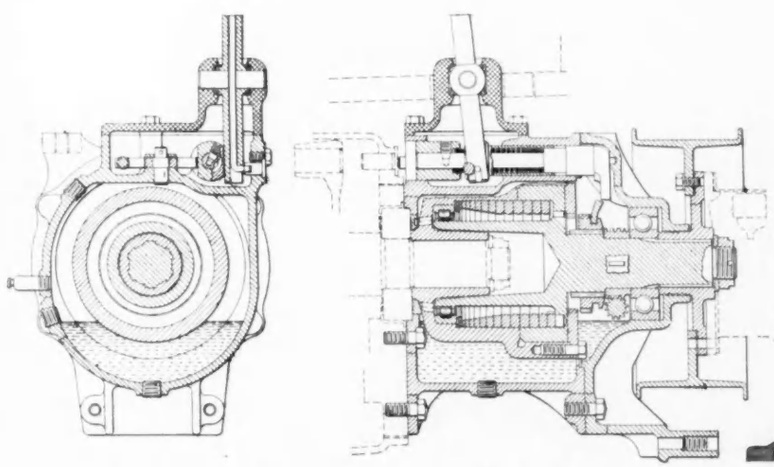


FIG. 16—FREE-WHEELING UNIT OF DUESENBERG CAR

low initial angle of the cam breaks the oil film and stops the spinning of the roller. H. E. Churchill, of the Studebaker research department, has found that the cold-weather characteristics of a cam are determined by the cam angle and that slippage will occur at a given angle and a given viscosity. Thus, in the old design utilizing a nest of four rollers, slippage would occur with the smallest roller at an angle of 8 deg. 45 min. when the viscosity was 60,000 sec. Saybolt, corresponding to a temperature of 35 deg. fahr. for the oil that was being used. On the largest roller, engaging at an angle of 3 deg. 40 min., slippage would occur at 5,000,000-sec. viscosity, corresponding to — 20 deg. fahr. for the same oil.

The manufacturers of the spring-clutch type claim that their units will operate satisfactorily with any of the usual oils at temperatures from −20 to +100 deg. fahr., and that any oil that is suitable for the transmission is suitable for the free-wheeling device.

The lubricant in the free-wheeling case generally is kept at a level above the bottom of the shell raceway, or above the low point of the pocket of the spring-clutch diameter. In the Warner transmissions with free-wheeling units, the oil thrown off from the low-reverse gear is caught by a trough which leads it to the free-wheeling case. The overflow from this case returns to the transmission case. Detroit transmissions have a main-shaft ball bearing with a ring anchored in the outer race, as shown in Fig. 12, which acts as a seal to prevent lubricants from passing through the bearing excessively, because of the pumping action of the helical gear, and keeps out foreign matter.

A collector ring is provided at the back of the Dodge shell to trap the oil thrown off from its rear face. Horizontal holes convey the lubricant to the raceway.

Recommendations for classification of free-wheeling lubricants are being considered jointly by Subcommittee XXVII of Committee D2 of the A.S.T.M. and the Lubricants Division of the S.A.E. Standards Committee.

Free-Wheeling Control

All of the American free-wheeling devices have provision for automatic locking when shifting into reverse, and the locking member usually is disengaged by a spring when the gearshift lever is returned to neutral. The Auburn, Duesenberg and the Nash 1060 and 1070 models have a separate shift lever immediately in front of the seat to lock the free-wheeling unit in or out. Most of the other cars are provided with a dash control in the form of a lever or push-and-pull knob. A customary notation on this knob is FREE-WHEELING—IN. In the case of Studebaker, the dash lever is connected to a tell-tale that appears behind a gage glass on the instrument board. The dash controls usually operate through a Bowden wire to a lever or fork actuating the free-wheeling unit. A very simple form of construction is used in the Chevrolet, as shown in Fig. 13.

Hudson and Essex have a very distinctive control by a movable ball at the head of the gearshift lever, as shown in Fig. 18. Raising or lowering the ball actuates a bell-crank, the short arm of which fits into the groove of the free-wheel locking gear. A spring connection between the bell-crank and the control wire allows independent actuation of the lock when going into reverse.

The various cars produced by the Chrysler Corp. have the free-wheeling control interconnected with the automatic vacuum clutch-control valve. This is intended to provide easy gearshifting and to obviate the necessity of a synchronizing mechanism by isolating the transmission at both ends. Recently the control has been modified to provide an intermediate position in which the automatic action of the clutch is cut out and free-wheeling is maintained. The addition of the vacuum clutch-control to Hudson and Essex cars, in addition to the present free-wheeling unit and synchronizing mechanism, has recently been announced. The control for the vacuum valve is located on the steering column and is not interconnected with the free-wheeling unit.

Warner Controls of Two Types

Fig. 19 is a horizontal section through the retractor mechanism of the Warner T-82 transmission. This retractor is at the left of the free-wheeling unit and at the level of its main shaft. The control mechanism of the T-81 transmission, shown in Fig. 9, is similar. When the reverse gear pushes the lock gear to the rear, by the key shown in Fig. 9, it also moves the shifter fork and compresses the spring that surrounds the inner rod, which is the upper one in the illustration. When the driver shifts from reverse back to neutral, the spring automatically unlocks the free-wheeling unit.

The rest of the mechanism in Fig. 19 takes care of putting the free-wheel device in locked or conventional position at the will of the driver by means of the instrument-board control. The transmission end of the Bowden wire is attached to the upper end of a vertical lever which has, below its ball fulcrum, a fork extension that connects with the bar between the two parallel rods, at the left in Fig. 19. The outer rod, the lower in the view, acts as an interlock to keep the fork in locked position until the control on the dash has again been retracted and the device automatically returns to the free-wheeling position.

To prevent shock loads on the lock gear and the mating internal gear in the shell, some of the manufacturers provide an interlock so that the clutch must be

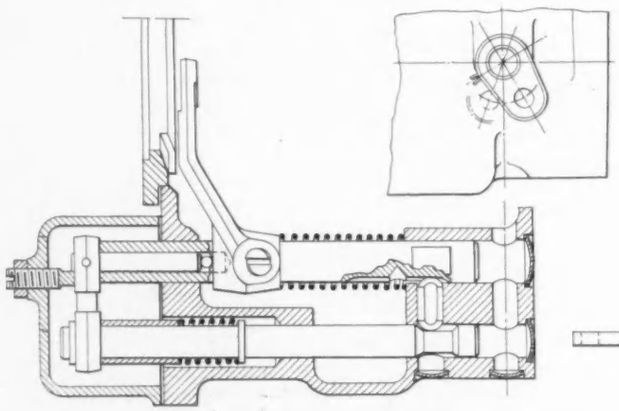


FIG. 19—CONTROL PARTS OF WARNER T-82 UNIT

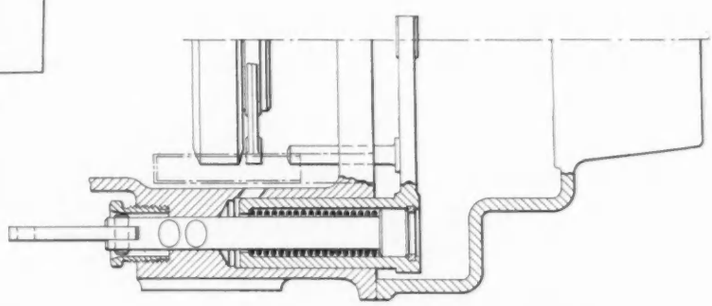


FIG. 20—CONTROL PARTS OF WARNER T-83 UNIT

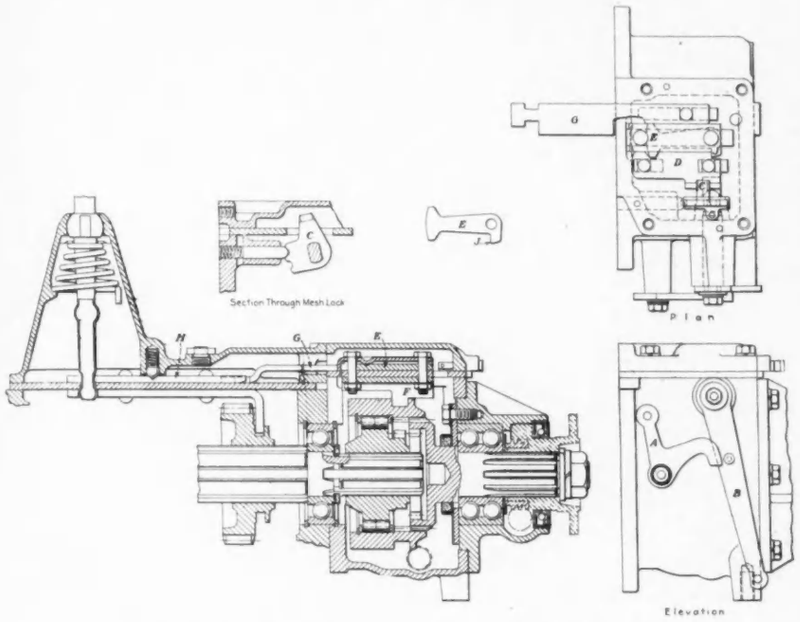


FIG. 21—CONTROL PARTS OF DETROIT SM-400-F UNIT

disengaged before attempting to lock out the roller ratchet. The small horizontal lever above the rear end of the free-wheeling case, shown in plan above the section in Fig. 19, is spring-connected to the clutch pedal. This small lever controls the rise and fall of a vertical plunger by a helical slot in the latter, and it drops into or rises from the slot on the upper side of the fork rod, shown at its right end. When the plunger is down in the slot, the operator must release the clutch pedal before he can move the locking fork in the direction of locking. It is not necessary to depress the clutch pedal for unlocking, because, if the free-wheel control on the instrument board has been returned to the free-wheeling position, the unit will unlock itself automatically the next time the free-wheel cam is relieved of the driving load. This occurs so smoothly that the driver does not feel it.

A somewhat different type of control is used in the Warner T-83 transmission, as will be seen from Fig. 20. It has the same automatic-return feature. The free-wheel locking fork has a long hub, the outside diameter of which fits in a counterbore in the rear face of the transmission case. The hub is bored and counterbored to accommodate a headed shift rod and a spring acting between the head of this rod and the bottom of the counterbore. The head of the rod stops against a snap-ring at the rear end of the hub bore, so that the headed rod carries the fork with it when it is pushed to the rear and thereby locks the free-wheeling unit. The shift rod is operated from the instrument board as in the other models. When the driver desires to return to free-wheeling, the control on the instrument board is placed in the free-wheeling position, pulling the shift rod forward and compressing the spring within the hub of the locking fork. The next time its teeth are relieved of their load, the lock gear automatically returns to free-wheeling position.

For locking up when reversing, the locking fork carries a shouldered pin which passes through the rear wall of the transmission case and contacts with a boss on the fork of the low-reverse gear of the transmission, as shown also for the Hudson in Fig. 10. Rearward movement of the transmission fork pushes the locking fork back into the locked position. The locking rod is held from moving by a spring detent, and the spring within the hub is compressed. This spring returns the lock fork to the free-wheeling position when the low-reverse gear is returned to neutral. This construction requires that the detent spring be stiff enough to withstand the retractor spring, and it has proved to work out satisfactorily. This transmission has an interlock similar to that on the T-82 to prevent locking the free-wheeling unit without throwing out the clutch.

Control of Detroit Unit

In the Detroit SM-400-F free-wheeling unit, shown in Fig. 21, depressing the clutch pedal moves the bell-crank lever *A* on the side of the free-wheeling case in a counter-clockwise direction, thus clearing the adjacent pin on the long lever *B*. The latter is connected with the instrument-board mechanism by a wire control. Movement of lever *B* transmits a corresponding movement to the short shift lever *C*, which is mounted on the same shaft as is *B*. This shifts plate *D*, which carries with it the interlock *E* and fork *F*, effecting the shift.

When going into reverse, bar *G*, which is connected with the transmission shifting plate *H*, is moved backward. As it moves, interlock *E* drops into the slot provided in bar *G*. When the shoulder on this bar is moved sufficiently, it shifts the interlock back and carries the fork *F* into the locking position. The lock gear is stationary while the rest of the free-wheeling unit slides backward. When the transmission shift lever is returned to neutral, interlock *E* is carried forward with bar *G* until point *J* on the interlock strikes plate *D*. The front end of the interlock is then thrown to the left to permit bar *G* to return to the position shown. The spring and detent shown retain the fork in the locked and unlocked positions.

In the Detroit SM-400-L construction, shown in Fig. 12, the shift to positive drive is accomplished in the same manner as described in the foregoing except that a spring, *K*, returns the shift fork to free-wheeling position when pressure of the gearshift lever is removed. When the transmission is shifted into reverse, bar *G* moves back as in the F type, pushing interlock *E* before it. When the transmission is returned to neutral, the pressure of bar *G* on the interlock is released, and the spring again forces the interlock and the shifting fork into free-wheeling position.

In the greater number of cases the free-wheeling unit does not move axially on the shaft, as it does in some of the Detroit units. These are reminiscent of the sliding unit inside the 1931 transmissions.

Bearings and Other Design Details

The shell of the free-wheeling unit in the Chrysler design, shown in Fig. 8, is piloted by a bearing that is located outside the overhung rear end of the transmission main shaft. The other bearing for the shell assembly is at the rear of the free-wheeling case. In the Warner transmission shown in Fig. 9, the free-wheeling shaft is guided at its forward end by a roller bearing inside the transmission main shaft and within its rear main bearings. The pilot bearing in the Hudson construction, Fig. 10, is located in approximately the same longitudinal position as in the Chrysler, but it is much smaller.

No pilot bearing is used in the various Detroit units.

A large double-row bearing is located at the end of the free-wheeling case, and the forward end in one type depends on the self-centering action of the rollers on the cam.

A roller bearing is located at the forward end of the inner sleeve in each of the spring-clutch designs except the Auburn-Twelve unit, which utilizes a plain bushing. A separate outer raceway is utilized in the Duesenberg construction (Fig. 16). The large-diameter clutching member of the L.G.S. device as now used on the Auburn Eight and Chevrolet (Fig. 12) is giving way to the more compact construction used on the Auburn Twelve and Duesenberg (Figs. 15 and 16).

Some comments on design contributed by two members of the Society are deemed of sufficient interest to record here. K. M. Wise, director of engineering of the Pierce-Arrow Motor Co., says:

> We are highly pleased with our present free-wheel unit, but it may not be amiss to recount some of the features that have given us concern in the past to indicate what must be guarded against in any design. We have had to learn how to build better transmission gears; and some of the earlier so-called transmission lubricants, which were designed as much for silencing as for lubrication, can no longer be used. A free-wheel unit of our type must have a lubricant that is not too viscous if it is to function properly at low temperatures. Improvements that have been made render the unit less sensitive to viscosity of the lubricant than formerly, so that no highly specialized lubricant need be employed with the present design.
>
> Excessive wear from continued free-wheeling has caused trouble with various designs, having been known to cause galling and eventual locking in one type. A roller clutch also can wear in such a way as to cause failure to drive.

P. L. Tenney, chief engineer of the Muncie Products Division of the General Motors Corp., writes:

> Experimental work that we have done and designs that we have analyzed show that attention has been concentrated on the holding power of the clutch and very little has been paid to other features of performance or to the economics of manufacture. I refer particularly to the effect of eccentricity and misalignment on the action of the free-wheel device and its influence upon the transmission. Evidently very little attention has been paid to this in the existing designs. For instance, several designs are in production in which the jaw clutch is placed at the far end of the splined transmission shaft in such a way that any eccentricity of the jaw-clutch parts, which are impossible to manufacture as designed without some permissible eccentricity, impose a very severe condition of misalignment upon the transmission shaft and the free-wheel unit.
>
> Most of the present designs are reasonably satisfactory as to roller contact and hardness and strength of parts, but we believe that a revision of design will be necessary to avoid the condition pointed out before real satisfaction can be expected.

A Recent English Design

Rejuvenation of free-wheeling activities by American manufacturers has spurred further developments in Europe, where this movement was at a low ebb. One of the latest designs is the Millam unit, Fig. 22, produced by Armstrong Whitworth & Co., of England. This is a heavy unit designed for commercial vehicles, which was based on the passenger-car type of the same company and has been tested out in service for a considerable period. The device is a self-contained unit, placed between the transmission and the rear axle. A separate oil-pump is provided to assure adequate lubrication. The device also incorporates a sprag and an "engine pick-up."

In this apparatus, the torque is transmitted between a driving member, which resembles an ordinary jaw clutch, and a surrounding drum. Two hard-steel rollers are disposed radially between each two jaws of the driving member. Driving pads or wedges are so placed between the periphery of the driver and the inside of the drum that the outer of each pair of rollers thrusts against the inclined inner face of the pad. The reaction to this thrust is transferred by the inner one of each pair of rollers to a central ring attached to the drum. Thus the heavy radial thrust and its reaction are taken wholly by the drum and its ring, and not between the drum and the driving member, as is usual.

A slight drag occurs between the driving pads and the drum when the device is called upon to operate. The driving member then tends to overrun the drum, and the rollers tend slightly to overtake the pads and roll along their inclined inner surfaces, automatically setting up a pressure between the drum and the central roller. Consequently the driving member is locked to the drum and cannot become free until the drum tends to overrun the driving member. When the pads gain on the rollers, the radial pressure automatically vanishes and the drum runs free of the driving member.

The sprag consists of a fixed ratchet and rotating pawl, slideably fitted in the drum. It is automatically put out of action when the free-wheel device is locked.

Fear has been expressed that, in case of engine stoppage while a commercial vehicle not fitted with a self-starter is free-wheeling, it would be necessary to stop the vehicle for the driver to restart engine. To overcome this objection, the Millam free-wheel incorporates a device by which the engine can be started by the vehicle, although the free-wheel is in action. This is accomplished through a cone clutch in the free-wheel housing, actuated by the free-wheel control lever. The clutch consists of a steel cone slideably attached to the driving member and working in a fabric-lined outer case attached to the drum, the whole running in oil.

Future Developments

The present trend toward simplification of control certainly is receiving a temporary setback by the addition of the various free-wheeling controls. In view of the two necessarily different conditions encountered, no objection can be raised to the usual dash control to lock out the free-wheeling. The clutch interconnection, necessitating clutch operation when endeavoring to lock out, is an added operating complication, and attempts are now being made to provide a speed synchronizing means for facilitating engagement and obviating possible shock loads.

I believe the immediate future will see the roller-ratchet type incorporate the maximum possible number of rollers in order to reduce the unit loading to the minimum. This will decrease wear and result in a mechanism less sensitive to the characteristics of the lubricant. Improved means will be provided to decrease spinning when free-wheeling. The cam should be self-centering, to avoid alignment difficulties and unbalanced engagement or disengagement.

A few years ago the No-Back device was used on several cars. It was a roller-ratchet device having one roller and a "fixed" cam which had a slight movement between locking and free positions. All of the elements of a free-wheeling clutch could be used to prevent a vehicle from backing down a grade, the only differences being in the relationship of the driving connections and anchoring to a stationary member. I see no reason why the present free-wheeling unit cannot be expanded to accomplish this function.

With the advent of jumbo tires, brake-drum diameters are coming into conflict with rim diameters and a repetition is now under way of the fight of a few years ago between these two elements in heavy-duty vehicles.

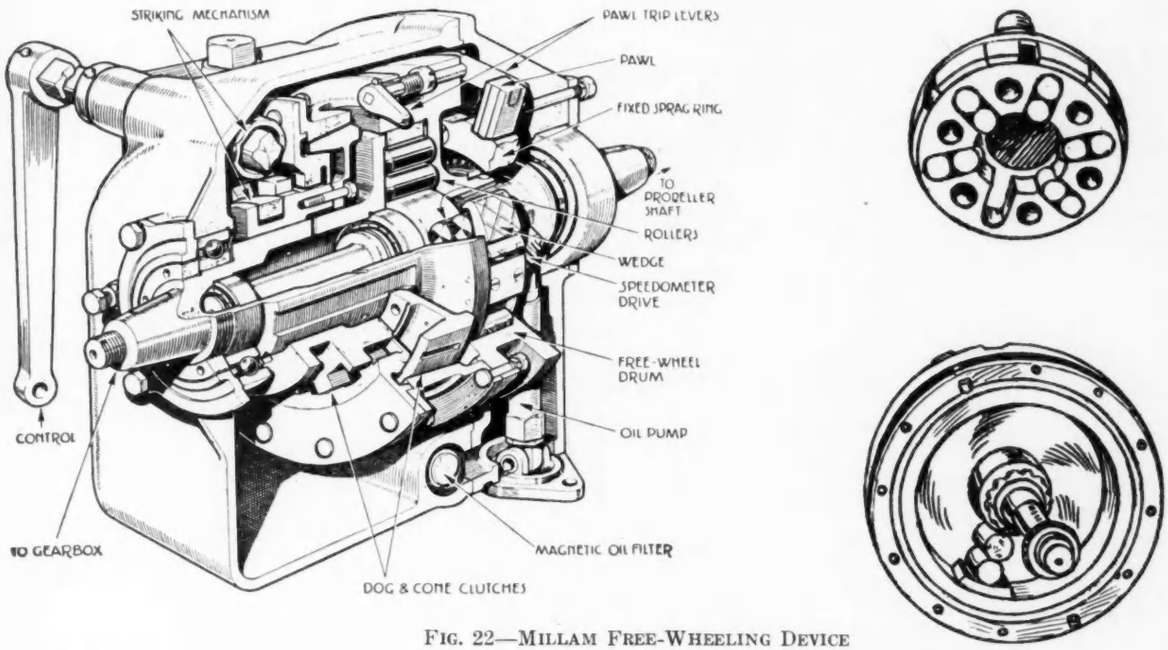


FIG. 22—MILLAM FREE-WHEELING DEVICE

The Wedge-Shaped Shoe, Shown in the Detail at the Right, Is Pressed Against the Drum by the Action of Two Rollers, the Reaction Being Taken by a Control Ring Which Is Rigid with the Drum

I believe that the service brake will have to be removed from the rear axle, for lack of sufficient space and to take unsprung weight away from the axle. Then the propeller-shaft brake will be the proper type to use, and it could be built into a unit with the free-wheeling device. It also favors true equalization.

I am of the opinion that a multiple-disc or plate unit can be built, all of which might be allowed to rotate, while the vehicle is being driven, from the drag of the plates. When braking, the "stationary" member of this unit would be engaged and brought to rest by a lock gear, after which, pressure on the plates would be applied in accordance with the braking demands. The picking-up of the "stationary" member would be easy, and could be facilitated by staggering the engaging teeth of the clutching member. This construction would avoid brake drag. The necessary elements of such a design are locking gears; brake unit, which will also function as a synchronizing clutch; means for preventing reverse rotation; and the various controls. The whole could be coordinated into one assembly to provide a free-wheeling unit, a speed-synchronizing device for the same, a sprag device and a service brake. The common usage of many of the elements involved should place the accomplishment of this within the ability of the designing engineer. The clutch or brake unit could incorporate a blower, which would of course be inactive when the entire unit is rotating but would come into increasing use as the relative motion between the two members became greater, to dissipate heat when the unit is used as a brake.

No free-wheeling unit has been adopted for heavy-duty vehicles in this Country, for reasons that probably are well known. The operating sequence of the mechanism proposed in the foregoing could be made such that, when the driver applies the brake while the vehicle is coasting, the clutch unit would first function as a synchronizing clutch so that the initial retardation of the vehicle would be obtained by speeding up the engine. Following this action, the clutch unit would become a brake, thus using both the engine and the brake for retardation. Timing control that can be inherent in the design would cause a light application of the brake to react merely against the engine, without use of the clutch unit as a brake. The vehicle would resume free-wheeling upon release of the brake pedal. The clutch unit would act as a brake in a real stop. Release of the pedal could allow the mechanism to reverse its sequence of operations either automatically or by means of manual control.

What we shall do in the far-distant future, when an infinitely-variable-ratio transmission is available and the Saurer method of utilizing the engine as a brake by transforming it into a compressor may be revived, is hard to tell. Many developments will have been wrought before that period arrives, and until then there is no doubt that free-wheeling will be an important factor in vehicle design and operation.

Fundamentals of Automotive Lubrication[1]

By H. C. Dickinson[2] and O. C. Bridgeman[3]

Annual Meeting Paper

SATISFACTORY performance of a lubricant depends upon characteristics of the lubricant, operating conditions and design of the device in which the lubricant is used. Applied lubrication requires a study of the relation among these factors in their effect upon performance.

The authors treat journal bearings, ball and roller bearings and gears. Equations are given for journal bearings operating under various conditions of design, lubrication, friction and heat dissipation.

The authors conclude that neither ZN/P nor PV alone is adequate as a measure of the power dissipated by a bearing, a composite relation involving both terms being required over a large part of the operating range. They show that each bearing has a minimum value of ZN/P below which it may get into the unstable region of thin-film lubrication and fail.

Temperature rise in the oil film was studied and the recommendation is made that the metallic parts should be cooled, as by the use of an efficient oil-cooler and a system that will provide effective heat transfer from the metallic parts to the cooled oil.

TRANSFORMATION of energy into useful work, in most of its manifold applications, involves a mechanism for power transmission, and this in turn introduces that formidable factor, power loss. In the transmission of electrical energy, the resistance of the conducting medium is the important factor in determining power loss. With mechanical energy, the power loss is determined by the friction between parts moving relatively to one another. Evidently, conservation of energy demands that friction be reduced to the minimum, and the prime requisite of a lubricant is that it shall perform this function of minimizing friction. Whether this function is performed satisfactorily depends upon the characteristics of the lubricant, the operating conditions and the design of the device in which the lubricant is being used. A serious study of the relation between these three factors in their effect upon performance is the major problem of applied lubrication.

In the utilization of mechanical energy, two main kinds of device are involved; namely, load-carrying devices and power-transmitting devices. From the automotive viewpoint, the two general types of load-carrying device of most importance are journal bearings and ball and roller bearings. While there are many devices for transmitting power mechanically, only positive power-transmission will be considered herein, and of the devices for this purpose gears are the most important. These three types of devices—journal bearings, ball bearings and gears, will be treated, in the order named, from the viewpoint of available information on the relation between lubricant characteristics, operating conditions and design factors, as the three independent variables, and some performance factor as the dependent variable.

The choice of the particular performance factor of interest depends somewhat upon the type of device under consideration. Thus, in the case of journal bearings, friction is the most important performance characteristic, since, under given operating conditions, this determines how much power will be dissipated in the bearing and hence how much heat must be removed per unit of time. In a well-designed journal bearing suitably protected from abrasive materials, wear and seizure are not of special consequence. They may, however, become of importance if other requirements necessitate a bearing that is poorly designed for the particular operating conditions, if the bearing is used under operating conditions differing widely from those for which it was designed, or if it is impracticable to provide suitable protection from abrasives. In the case of ball and roller bearings, friction is not of as much consequence as with journal bearings, since rolling is involved rather than sliding. Also, wear and seizure are not highly important if these types of bearings are correctly designed and are kept free from abrasives, although, because of the higher loads per unit area which are involved, these factors are of more consequence than with journal bearings. In the case of gears, where the loads per unit area of contact are frequently very high, wear and seizure become the most important performance characteristics, and friction measurements become of little significance except in relation to the other two factors.

Journal Bearings

A journal bearing consists of a cylindrical shell in which rotates a cylindrical shaft or journal. The shell may be stationary or may be rotating, as in the case of reciprocating mechanisms. The general principles are the same in the two cases, the difference involving design factors and methods of lubrication. With a well-designed bearing, suitably adapted for the particular operating conditions, there will be a continuous film of lubricant between the bearing surfaces. As the journal rotates, the film will be sheared, and at any given speed the friction will be directly proportional to the viscosity of the oil at the temperature existing in the oil film. Likewise, for any given viscosity of oil, the friction will be directly proportional to the speed of rotation, for the resistance is proportional to the rate of shear. Accordingly, in the general case of a concentric bearing, the friction will be proportional to the product of the viscosity and the speed, or

$$F = k_1 ZN \qquad (1)$$

where

F = friction
k_1 = a dimensional constant
N = speed in revolutions per minute
Z = absolute viscosity

As long as the film is sufficiently thick, the load on the bearing will have practically no effect on the friction, except insofar as the viscosity increases with pressure, and this is of minor importance with thick films. Equation (1) might be used as a definition of a well-designed bearing, which in that case would be of such length with respect to diameter and of such clearance that, under the given speed of operation, the friction would be independent of the load.

[1] Publication approved by the Director of the Bureau of Standards, Department of Commerce.

[2] M.S.A.E.—Chief, heat and power division, Bureau of Standards, City of Washington.

[3] S.M.S.A.E.—Chief, lubrication and liquid-fuels section, Bureau of Standards, City of Washington.

For many purposes, it is convenient to define a coefficient of friction in terms of the friction and load, according to the relation

$$f = k_2 (F/P) \qquad (2)$$

where

F = friction, as defined by Equation (1)
f = coefficient of friction
k_2 = a dimensional constant
P = load per unit of projected area

Combining Equations (1) and (2) to eliminate F there results

$$f = k_3 (ZN/P) \qquad (3)$$

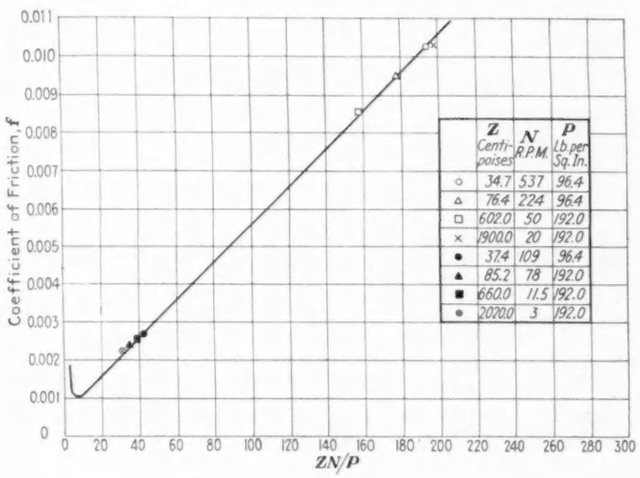
where $k_3 = k_1 k_2$ and depends upon the units employed and upon the dimensions of the bearing and journal.

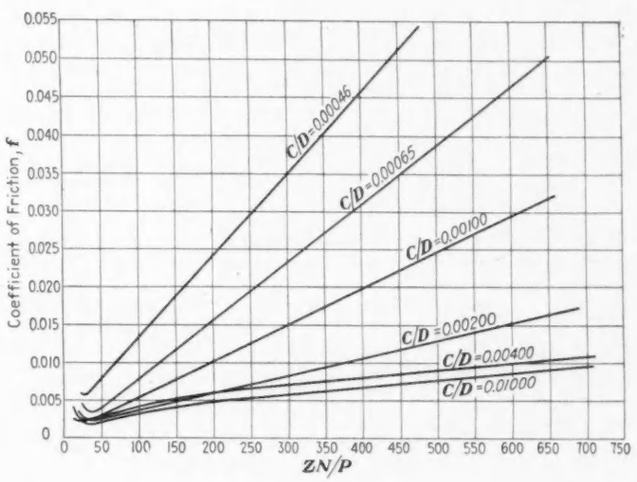
[4] See *Transactions of the American Society of Mechanical Engineers*, APM-51-15, p. 161, 1929.

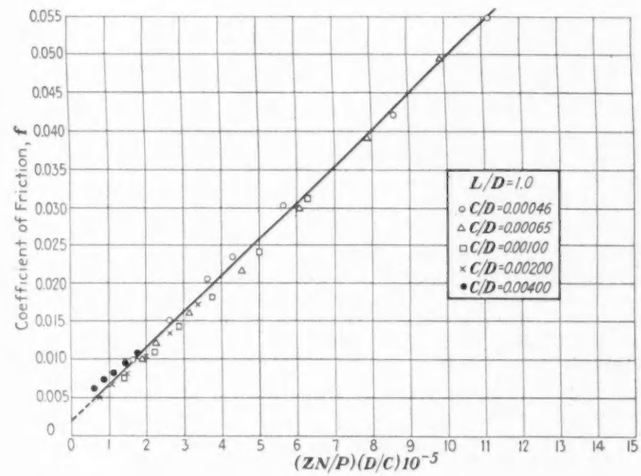
For an ideal bearing of given dimensions, which remains geometrically perfect under all operating conditions, Equations (3) purports to be a general relation between the viscosity of the oil and the operating variables (speed and load). It is restricted by the conditions that the oil film is sufficiently thick, that the journal runs concentrically in the bearing and that the end leakage of the oil from the bearing is negligible. In practice, owing to end leakage and lack of concentricity, the coefficient of friction as given by Equation (3) will be too low and it is necessary to add another term, a, which will be constant for any given dimensions of bearing. Equation (3) therefore becomes

$$f = a + k_3 (ZN/P) \qquad (4)$$

Experimental proof of Equation (4) is given in Fig. 1, using data obtained by McKee[4]. Over wide ranges of the individual variables, a straight line reproduces the







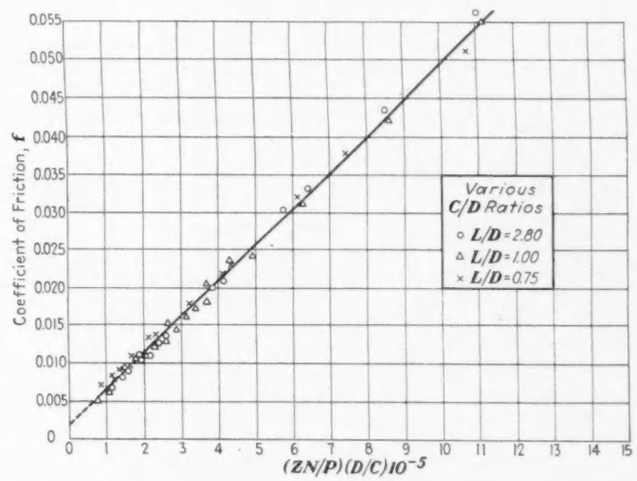


FIG. 1—CURVE FROM EXPERIMENTAL DATA WHICH PROVIDE PROOF OF EQUATION (4)

Over Wide Ranges of Individual Variables a Straight Line Reproduces the Data within Experimental Error Down to Low Values of ZN/P, in Which Region the Coefficient of Friction Begins To Rise with Further Decrease in ZN/P. In This Region a Transition from Thick-Film to Thin-Film Lubrication Occurs

FIG. 2—CURVES SHOWING RELATION BETWEEN FRICTION AND CLEARANCE

The Coefficient of Friction Is Reduced for the Same Value of ZN/P As the Clearance Becomes Smaller in Bearings of the Same Length and Diameter. Division of the ZN/P Valves by the Clearance Brings All the Lines into Accord

FIG. 3—EXAMPLE OF AGREEMENT BETWEEN EXPERIMENTAL DATA AND EQUATION (5) FOR "SQUARE" BEARINGS

FIG. 4—CURVE SHOWING RELATION BETWEEN COEFFICIENT OF FRICTION AND ZN/P

From Experimental Data Showing That, for Bearings in Which the Length-Diameter Ratio Is Greater than 0.75, the Relation Is Independent of the L/D Ratio

data within experimental error down to low values of ZN/P, in which region the coefficient of friction begins to rise with further decrease in ZN/P. In this region, where departure from linearity occurs, there is a transition from thick-film to thin-film lubrication.

In the case of a bearing of given dimensions operating in the régime of thick-film lubrication, Equation (4) gives information regarding the change in coefficient of friction with changes in the individual variables, Z, N and P. It also indicates that these individual variables can be assigned different values in such a way that ZN/P, and hence f, remains constant. Thus, if an oil of given viscosity at the operating temperature is used, a doubling in speed permits a doubling in the load carried. Similarly, if the load is doubled at constant speed, the viscosity should be doubled to compensate.

The values given in Fig. 1 are applicable only to a bearing of given dimensions. If other bearings of the same length and diameter, but with various clearances, are considered, a family of curves will be obtained as illustrated in Fig. 2. As the clearance becomes smaller, a lower coefficient of friction is obtained for the same value of ZN/P. Analysis[5] of the curves indicated that division of the ZN/P values by the clearance brings all the lines into accord. Therefore, dividing ZN/P term in Equation (4) by C/D, the ratio of clearance to diameter, there results

$$f = a + K\,[(ZN/P)\ (D/C)] \qquad (5)$$

The ratio C/D is used rather than C, since it makes the resulting equation applicable to bearings of various diameters.

An example of the agreement between the experimental data of McKee and Equation (5) is shown in Fig. 3 for "square" bearings.

There is one other design factor in connection with bearings; namely, the ratio of the length to the diameter. For bearings in which the L/D ratio is greater than 0.75, it has been found experimentally[5] that the relation between coefficient of friction and ZN/P is independent of L/D, within experimental error. This is shown in Fig. 4, where L/D varies from 0.75 to 2.8. Accordingly Equation (5) purports to be a general relation between coefficient of friction, oil viscosity and design and operating factors for all usual values of C/D and for L/D greater than 0.75 in the régime of thick-film lubrication, and hence is applicable to most automotive problems involving journal bearings.

The usefulness of Equation (5) can be illustrated by specific cases. Consider a bearing operating under such conditions that $ZN/P = 50$. If the speed is decreased at constant load with a given oil, the coefficient of friction and hence the friction becomes less; the temperature of the bearing decreases with resultant increase in the viscosity of the oil and there is a tendency to increase the value of ZN/P. In other words, there is a restoring tendency away from the régime of thin-film lubrication. The same reasoning applies if the viscosity of the oil is decreased at constant speed and load. When changes in the load are made, the situation is somewhat different. Combine Equations (2) and (5) and there results

$$F = mP + nZN \qquad (6)$$

where $m = a/k_2$ and $n = (K/k_2)\,(D/C)$. If the load is increased at constant speed with a given oil, the friction increases as indicated by Equation (6). This in turn tends to increase the oil temperature, with resultant decrease in the viscosity of the oil. Reference to Equation (5) indicates that an increase in load and a resultant decrease in oil viscosity both tend to lower the value of ZN/P and hence force the bearing toward the régime of thin-film lubrication. Thus, while there is an automatic restoring mechanism for changes in speed and viscosity at constant load, not only is there no such restoring tendency for changes in load, but there is an actual tendency toward a region of instability. This tendency is offset by the fact that, as the temperature of the bearing increases, more heat is removed from it. Also, as the viscosity of the oil decreases, the friction and therefore the temperature tends to decrease. Hence, if the bearing is operating in a region not too close to the unstable region, a new equilibrium state will be reached before actual instability occurs. It is therefore particularly desirable that the ZN/P value for a given bearing be sufficiently high so that any increase in the load, at the lowest speed and oil viscosity used, will not place the bearing in the régime of thin-film lubrication.

In using Equation (5) for designing bearings, all of the operating and design variables, with the exception of the oil viscosity, can be assigned values directly. The assignment of a value to the viscosity depends upon the temperature in the oil film, and this in turn is dependent upon the power dissipated in the bearing, the rate of heat conduction away from the bearing and the difference in temperature between the oil in the film and the metallic surfaces of the bearing.

Consider first the power of H dissipated in a bearing:

$$H = \pi D^2 NPfL \qquad (7)$$

Combining Equations (5) and (7) and rearranging there results

$$H = PN\,[a + K\,(ZN/P)\ (D/C)]\,\pi LD^2 \qquad (8)$$

From the lines in Figs. 3 and 4, which are based on the extensive experimental work of McKee, it is seen that the constant a has the value 0.002 and the constant K is equal to 473×10^{-10}. Substituting these values in Equation (8) it becomes

$$H = PN\,[0.002 + 473 \times 10^{-10}\,(ZN/P)\ (D/C)]\,\pi LD^2 \qquad (9)$$

Equation (9) is restricted to values of ZN/P which are not too close to the minimum point on the f-ZN/P curve. It is not possible to state a limiting value, since the minimum point varies with the bearing materials, the amount of running-in, and, to some extent, with the values of speed and load. Also, in the region near the minimum point, the values of f found experimentally are frequently lower than those given by Equation (9), presumably as a result of deformation.

In the last few years, there has been considerable discussion as to whether the power dissipated in a bearing was given by ZN/P or by PV, where V is the circumferential velocity and equals πDN. Equation (9) indicates that, while ZN/P may be used over a part of the operating range and PN, or PV, over another part of the operating range, neither expression alone is adequate to cover the power dissipated over the entire operating range. To illustrate this point, consider a bearing in which $L = 1$, $D = 1$ and $D/C = 1000$. Equation (9) can then be written

$$H = \pi PN\,[0.002 + 473 \times 10^{-7}\,(ZN/P)] \qquad (10)$$

If $ZN/P = 1000$, the term 0.002 is small as compared with the ZN/P term, and the power dissipated is proportional to ZN^2. If Equation (10) is still applicable and $ZN/P = 5$, the ZN/P term is small as compared with 0.002, and the power dissipated is proportional to PN or to PV. When $ZN/P = 42$, the two terms become of equal magnitude. Accordingly, over a large part of the operating range, the power dissipated in a journal bearing is given by a composite relation involving both ZN/P and PV. Which of these two terms is of greater importance in determining H under any given set of conditions is dependent upon the value of ZN/P.

Equation (9) is of major importance in showing how the power dissipated, and hence the heat generated, varies with design features and with operating conditions. Thus, the heat generated varies inversely with the diam-

[5] See *Transactions of the American Society of Mechanical Engineers*, APM-51-15, p. 161, 1929.

etral clearance, and directly with the length of the bearing and with the viscosity of the oil. The effects of variations in speed, load and diameter of journal are somewhat more involved but can readily be evaluated from the equation.

As heat is generated in a bearing under any given set of operating conditions, the temperature of the oil tends to rise, with resultant decrease in oil viscosity. This in turn decreases the value of ZN/P. It was shown earlier in this paper that there is a minimum value of ZN/P for any bearing below which it is not safe to go, or otherwise the bearing may get into the unstable region of thin-film lubrication and fail. It is usually advisable to choose a value of ZN/P considerably on the safe side of this unstable region, and it is believed that this value should not be lower than 20 for the lowest speed and maximum load likely to be encountered. Choice of the minimum value of ZN/P for the lowest speed N and the maximum load P immediately determines the lowest permissible viscosity Z of the oil to keep ZN/P above the chosen value. If an oil of known viscosity at 130 or 210 deg. fahr. and a known temperature coefficient of viscosity (viscosity index) is being used in the bearing, establishment of the lowest permissible viscosity determines the maximum temperatures to which the oil can be heated in the bearing. In Table 1 the maximum allowable bearing temperatures are listed for three chosen values of ZN/P, when using both an S.A.E.-30 and an S.A.E.-50 oil in a bearing under various conditions of speed and load. For these calculations average values of viscosity and viscosity index have been used for each S.A.E. grade. Although a minimum ZN/P value of not lower than 20 is recommended, many bearings do operate in practice with ZN/P values of 10 or even lower, and Table 1 is intended to apply to such bearings. It is seen that the maximum allowable temperatures are very low, and it is of considerable interest that there is so little difference in permissible temperatures between the S.A.E.-30 and S.A.E.-50 grades. Permissible temperatures for $ZN/P = 20$ would be still lower.

It is to be noted that the maximum allowable bearing temperatures increase as lower values of ZN/P are chosen. While values of ZN/P given in Table 1 are considered to be too low, choice of these low values does permit less efficient cooling of the bearings. On the other hand, the lower the value of ZN/P is, the less margin of safety there is and the greater is the danger that a sudden abnormal increase in load may place the bearing in the unstable region. A compromise is therefore necessary between the margin of safety in ZN/P, the heat generated in the bearing and the efficiency of cooling of the bearing.

In the preceding discussion it has been assumed that the oil temperature was uniform throughout the oil film. That this is true for all practical purposes can readily be shown. Consider a thin section in the oil film of width dx and of unit area parallel to the length of the bearing. If θ is used to represent temperature and K to represent the thermal conductivity of the oil, then the following conditions hold:

$$\text{Rate of heat flow out of section} = -K\left[\frac{d\theta}{dx} + \frac{d^2\theta}{dx^2}\,dx\right]$$

$$\text{Rate of heat flow into section} = -K\frac{d\theta}{dx}$$

$$\text{Net rate of heat flow out of section} = -K\frac{d^2\theta}{dx^2}\,dx$$

To establish equilibrium, the net rate of heat flow out of the section must be equal to heat $h\,dx$ generated in this section per unit time, where h is the heat generated per unit volume and per unit time. Therefore

$$h = -K\frac{d^2\theta}{dx^2} \tag{11}$$

Since $h = \frac{ZV^2}{l^2}$ for viscous flow, l being the thickness of the oil film, this equation can be integrated keeping h constant, for Z is essentially constant at all points across the oil film and the velocity gradient V/l can be assumed constant. Integrating

$$\theta = -\frac{hx^2}{2K} + C_1x + C_2 \tag{12}$$

where C_1 and C_2 are integration constants. Let the temperature at the bearing and journal walls be θ_1 and θ_2, respectively.

Then when $x = 0$, $\theta = \theta_1$, and when $x = l$, $\theta = \theta_2$. Accordingly,

$$\left.\begin{aligned} x &= 0, \quad C_2 = \theta_1 \\ x &= l, \quad C_1 = \frac{\theta_2 - \theta_1}{l} + \frac{hl}{2K} \end{aligned}\right\} \tag{13}$$

After substituting these values in Equation (12) it becomes

$$\theta = \theta_1 + \left[\frac{\theta_2 - \theta_1}{l} + \frac{hl}{2K}\right]x - \frac{h}{2K} - x^2 \tag{14}$$

Equation (14) gives the temperature θ at any point x in the oil film in terms of the thickness of the film and the temperature at the bearing and journal walls. To

TABLE 1—MAXIMUM ALLOWABLE BEARING TEMPERATURE, DEG. FAHR.[a]

P Lb. per Sq. In.	N R.P.M.	Assuming Critical $ZN/P = 10$ S.A.E. 30	Assuming Critical $ZN/P = 10$ S.A.E. 50	Assuming Critical $ZN/P = 5$ S.A.E. 30	Assuming Critical $ZN/P = 5$ S.A.E. 50	Assuming Critical $ZN/P = 2$ S.A.E. 30	Assuming Critical $ZN/P = 2$ S.A.E. 50
1,000	1,000	193	235	240	285	320	373
1,000	2,000	240	285	297	348	420	485
1,000	3,000	275	320	348	420	500	600
2,000	1,000	158	190	193	235	257	300
2,000	2,000	193	235	240	285	320	373
2,000	3,000	220	260	275	320	373	435
3,000	1,000	145	175	172	205	226	268
3,000	2,000	172	205	210	254	280	330
3,000	3,000	193	235	240	285	320	373
4,000	1,000	132	162	158	190	205	247
4,000	2,000	158	190	193	235	257	300
4,000	3,000	180	215	220	260	292	338
5,000	1,000	125	155	150	180	193	235
5,000	2,000	150	180	183	220	240	285
5,000	3,000	170	203	203	245	275	320

[a] These temperatures are computed from average values of viscosity.

find the maximum temperature it is necessary to differentiate θ with respect to x and equate the resulting equation to zero. Following this procedure

$$\left.\begin{aligned} \frac{d\theta}{dx} &= \frac{\theta_2 - \theta_1}{l} + \frac{hl}{2K} - \frac{hx}{K} = 0 \\ \text{or} \qquad x &= \frac{l}{2} + K\left(\frac{\theta_2 - \theta_1}{hl}\right) \end{aligned}\right\} \quad (15)$$

Therefore

$$\triangle\theta = \theta_{max} - \theta_1 = \frac{hl^2}{8K} + \frac{\theta_2 - \theta_1}{2}\left[1 + K\frac{(\theta_2 - \theta_1)}{hl^2}\right] \quad (16)$$

where $\triangle\theta$ is the maximum increase in temperature in the oil film above that of the bearing wall at a point around the bearing where the film thickness is l. Equation (16) holds regardless of whether the oil in the film is moving or is stationary. It is restricted, however, to a certain condition in which the temperature is uniform around the bearing wall and any temperature rise in the film is caused by viscous shear. If the temperatures at the bearing and journal walls are equal ($\theta_1 = \theta_2$), then Equation (16) becomes

$$\triangle\theta = \frac{hl^2}{8K} \quad (17)$$

To use Equations (16) and (17) for the computation of $\triangle\theta$, it is necessary to know the rate of heat generation per unit volume. According to the laws of viscous shear, the rate at which heat is generated per unit area is equal to ZV^2/l, while ZV^2/l^2 is the rate at which heat is generated per unit volume. Therefore

$$h = \frac{ZV^2}{l^2} \quad (18)$$

Combining Equations (17) and (18), there results

$$\triangle\theta = \frac{ZV^2}{8K} = \frac{\pi^2 D^2 N^2 Z}{8K} \quad (19)$$

since $V = \pi DN$. Equation (19) is applicable to the case where the temperatures of the bearing and journal walls are equal. A similar relation for the general case can be obtained from Equations (16) and (18). It is of interest that the temperature rise is independent of the oil-film thickness.

If D is expressed in inches, N in revolutions per minute, Z in centipoises, K in ergs per second per centimeter per degree centigrade, and $\triangle\theta$ in degrees centigrade, then it is necessary to introduce a dimensional constant equal to 1.792×10^{-5}. Accordingly, Equation (19) becomes

$$\triangle\theta = \frac{1.792 \times 10^{-5}\,\pi^2 D^2 N^2 Z}{8K} \quad (19a)$$

Equation (19) is based on the assumption that the temperature rise in the oil results solely from viscous shear. In this case, the coefficient of friction f is given by the relation

$$f = 473 \times 10^{-10}\,[(ZN/P)/(D/C)]$$

and by substitution in Equation (19a) there results

$$\triangle\theta = 467\,PNDfC/K \quad (20)$$

The temperature rise in an actual bearing would be expected to be higher than that indicated on the basis of viscous shear, since the coefficients of friction found for actual bearings are higher than those predicted from viscosity considerations alone. Assuming that the ratio of the actual to the theoretical temperature rise is equal to the ratio of the actual to the theoretical rate of heat generation, then use of the measured coefficient of friction in Equation (20) rather than the theoretical value should give a very close estimate of the actual temperature rise in the oil film.

Equation (20) will be used to compute values of the temperature rise $\triangle\theta$ in a few typical cases. For this purpose, K will be taken equal to 1.3×10^4, which is an average value for a lubricating oil and varies only slightly with viscosity. The values obtained are listed in Table 2. The small temperature rises indicate that the heat generated is removed from the oil film to the metallic parts of the bearing at a very rapid rate. Hence the main problem in cooling a bearing is to remove heat from the metallic parts rather than from the oil film. Circulation of large quantities of hot oil through the bearings in the normal way is not the answer to the cooling problem. A more rational solution consists in the use of an efficient oil-cooler and a design of oil system which will not only furnish sufficient oil for lubrication but will also provide effective heat transfer from the metal parts to the cooled oil. Such a system could be designed that would keep oil temperatures in the film much below the values commonly found at present. As pointed out previously, values of ZN/P close to the danger zone are necessary with existing practice, since the permissible oil temperatures are higher in this region. Use of a more efficient system for cooling the oil and the bearings would therefore permit the designing of bearings for operation at higher and safer values of ZN/P.

TABLE 2—TYPICAL VALUES FOR TEMPERATURE RISE IN OIL FILM FOR $Z = 10$ CENTIPOISES, $P = 500$ LB. PER SQ. IN. AND $D/C = 1000$, C BEING CLEARANCE IN INCHES

D In.	ZN/P	N R.P.M.	f	$\triangle\theta$ Deg. Cent.
1	20	1,000	0.0030	0.05
1	40	2,000	0.0039	0.14
1	60	3,000	0.0048	0.26
2	20	1,000	0.0030	0.22
2	40	2,000	0.0039	0.56
2	60	3,000	0.0048	1.0
3	20	1,000	0.0030	0.48
3	40	2,000	0.0039	1.2
3	60	3,000	0.0048	2.3

The present treatment of journal bearings has been based on the information obtained at the Bureau of Standards in the last few years and is confined entirely to the régime of thick-film lubrication. Work is now actively in progress on the investigation of journal-bearing lubrication in the intermediate range between thick and thin-film lubrication.

Ball and Roller Bearings

The main characteristic of a journal bearing is the sliding of two surfaces relatively to each other, whereas, in the case of ball and roller bearings, rolling of one surface on another is largely involved. Since rolling is the main characteristic of "antifriction" bearings, it is found that the coefficient of friction obtained with this type of bearing is low. Thus, for radial ball and cylindrical roller bearings, f is about 0.001, and for thrust bearings, f is about 0.003, as compared with values from 0.002 to 0.005 for a well-designed journal bearing. These small differences in f are of little consequence, but it is of interest that, while the value of f changes considerably with operating conditions in the case of journal bearings, the coefficient of friction for an antifriction bearing changes only slightly over the usual range of operating conditions. Recent design practice in machinery, however, has tended toward the use of this type of bearing under high speeds and loads without material increases in size of bearing and it would be expected that under these conditions the coefficient of friction of ball and roller bearings might change with changes in the operating variables and this factor might become of considerable interest. Unfortunately, little

(*Concluded on p. 304*)

Ignition Quality of Diesel Fuels as Expressed in Cetene Numbers

Semi-Annual Meeting Paper

By G. D. Boerlage[1] and J. J. Broeze[2]

THE paper is an account of some further experiences and views on the subject of ignition in oil engines, which, with the advent of the high-speed Diesel engine, has received much attention of late. The Delft Engine Laboratory of the Royal Dutch Shell Group has been working on oil-engine fuels for nearly four years, with the aid and assistance of all the group's chemical and physical specialists. It had already been working on ignition problems before the high-speed engine was introduced, these problems belonging to a continuous series. Though ignition problems have become of paramount importance in present-day high-speed engines, they have always played a certain rôle with certain fuels also in low-speed engines which were not sufficiently recognized.

After discussing the three phases of combustion in compression-ignition engines, the authors analyze the fuel characteristics relating to ignition, describe engine tests for rating fuels and state that it was found expedient to base all ignition data obtained on reference fuels, for which, on theoretical considerations, cetene and mesitylene were suggested. This proved to be very effective, as the former substance ignites most readily in a normal engine, the latter not at all.

Fuel behavior in four different engines was studied to ascertain the degree of correlation, and charts are presented to illustrate the results obtained. The ignition quality and thermal stability of fuels are discussed in some detail, and a fuel formula to serve for estimating the ignition quality of straight distillate fuel oils from natural crudes is presented.

The general subject of ignition is considered, including nuclear ignition, and several special cases of curious ignition phenomena with regard to ignition problems are cited. In the general remarks on Diesel-engine design, the phenomena that are presumed to take place within the engine are reviewed, desiderata that are to be considered are stated, and an endeavor is made to combine these two groups so as to obtain guiding principles for design.

COMBUSTION in compression-ignition engines can, thanks to H. R. Ricardo[3] and others, be mainly divided into three phases as shown in the left diagram of Fig. 1, that is, Phase 1, delay period or ignition lag without perceptible rise of pressure; Phase 2, rapid combustion of the fuel that had already been injected; and Phase 3, controlled combustion of fuel injection after the end of Phase 2. This division is in the main correct, but it needs some amendment.

More recent results have shown that it is not altogether justifiable to call Phase 1—the delay period—the period without perceptible pressure rise, as, during long delay periods and especially in some engines, among others those with high turbulence, perceptible pressure rises do occur, during which even a high percentage of the total heat is sometimes developed, as indicated in the central diagram in Fig. 1. It would seem as if this combustion process differs from that with a prominent Phase 2, when it comes to extremes.

Ricardo has supposed that combustion in Phase 2 originates from a nucleus and he has shown that it can be controlled by turbulence, like combustion in a spark-ignition engine. Up to now little study has been made of this control, so that combustion in Phase 2 is practically uncontrolled and often very rough. In some cases, however, combustion in Phase 2 does not develop strongly enough to "catch up" the injection at the nozzle, which depends, among other factors, on flame temperature and turbulence in coordination. If this slow combustion takes place, there is no actual Phase 3 so far as good control is concerned. As regards the nuclear nature of ignition, this is referred to later in the paper and strong arguments are brought forward in support of this theory.

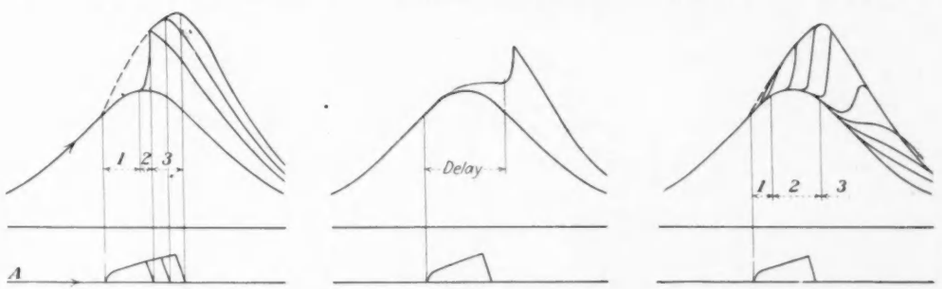


FIG. 1—COMBUSTION DIAGRAMS

Phases of Combustion.—Phase 1 Indicates the Delay Period; Phase 2, Uncontrolled Combustion; and Phase 3, Controlled Combustion. The Needle-Valve Diagram Is Shown at *A*

Perceptible Pressure Rise During Delay, the Needle-Valve Diagram Being Shown at *A*

Influence of Delay Period on Engine Behavior. — Smooth Combustion Is Indicated at 1; Increasing Roughness at 2; and Smooth but Incomplete Combustion, at 3

Keeping to the now usual type of engine to which Phases 1, 2 and 3 apply, the following possibilities for Phases 1 and 2 have been found to exist[4], as illustrated in the diagram at the right in Fig. 1.

(1) The delay period is short. The fuel, igniting practically as it enters, develops a gradual

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[3] See The High-Speed Internal-Combustion Engine, by H. R. Ricardo.

[4] See *Engineering*, Nov. 13, 1931, p. 603; Dec. 4, 1931, p. 687; and Dec. 18, 1931, p. 755.

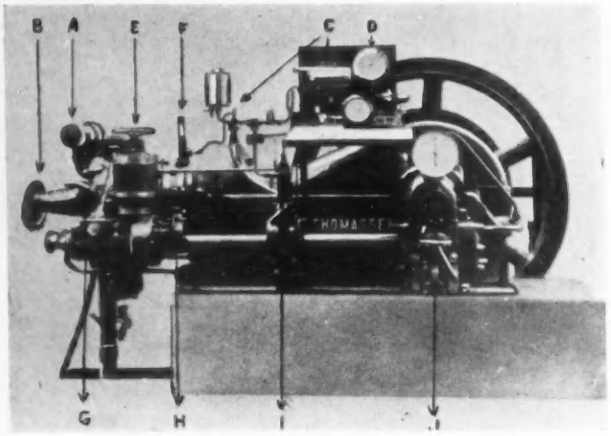


FIG. 2—THOMASSEN-DELFT FUEL-TESTING ENGINE

Locations of the Principal Features Are Indicated as Follows: *A*, Throttle Valve; *B*, Variable-Compression Plug; *C*, Fuel-Feed System; *D*, Hydraulic Brake-Load Meter; *E*, Governor for Variable Speed; *F*, Cooling-Water Outlet; *G*, Starting Valve-Gear, *H*, Piston and Valve-Gear Lubricator; *I*, Fuel-Pump, and *J*, Circulating Oil-Pump

pressure rise. The engines are smooth-running.

(2) An increase in delay period. More and more fuel accumulating during the delay period results in the building up of uncontrolled fierce combustion during which practically all the fuel already injected is consumed. Roughness of engines results, increasing with delay.

(3) A further increase in delay period. The circumstances that have led to the long delay retard and eventually prevent the formation of a rapidly burning flame. Decreasing roughness follows, then smoke and, finally, misfiring.

It is thereby seen that the delay period, which is a function of engine conditions and fuel characteristics, governs ignition and roughness. It governs also that kind of incomplete combustion intimately related to blue smoke, offensive acrid exhaust gases and, in some cases, very obstinate cases of piston gumming.[5]

[5] See *Journal of the Institute of Petroleum Technologists*, October, 1930, p. 678.

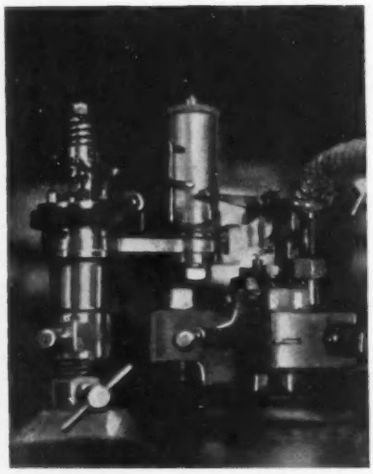


FIG. 3—INDICATOR ARRANGEMENT ON THE THOMASSEN-DELFT FUEL-TESTING ENGINE

Fuel Characteristics Relating to Ignition

The fuel characteristics relating to ignition can be classified as physical and as chemical characteristics.

(1) Physical characteristics are such as relate only to the physical preparatory processes that the fuels undergo in injection and evaporation. In the injection process, including atomization, the variables are viscosity, surface tension and specific gravity. In the evaporation process, there is volatility in addition to the other three properties so far as these influence the fineness of atomization.

(2) Chemical characteristics are those which determine the "reactivity" of the fuel, which mainly governs the delay period. This reactivity, we have found, should not be confused with the chemical reactivity against oxygen, at lower temperatures, which is well known to the chemists, but is related to the thermal stability of the fuel, as will be explained later.

Within wide limits the influence of the physical constants of the fuel on its preparation for ignition may be disregarded; or, as in the case of high viscosity, may be corrected by applying heat or adjusting the atomizer. Thus, for ignition in oil engines, the chemical characteristics are, from the fuel viewpoint, the more important. Generally speaking, this problem is very much in line with gasoline detonation problems. Between gasoline engines and oil engines, however, the situation is fundamentally different, as the efficiency of gasoline engines can principally be improved by the use of fuels of special ignition quality, whereas the efficiency of oil engines cannot. Therefore there is much reason to expect the development of the "omnivorous" oil engine that will be indifferent to fuels of special ignition quality, in contrast with the selective gasoline engine asking for "premium fuels."

Engine Tests for Rating Fuels

Engine tests were found necessary for a study of ignition problems, since, although numerous analytical data about fuels were at the authors' disposal, these were not readily convertible to enable a practical judgment to be formed about the suitability of these fuels as Diesel fuels.

Having at our disposal a series of engines of different types, the direct-injection engine seemed to us the

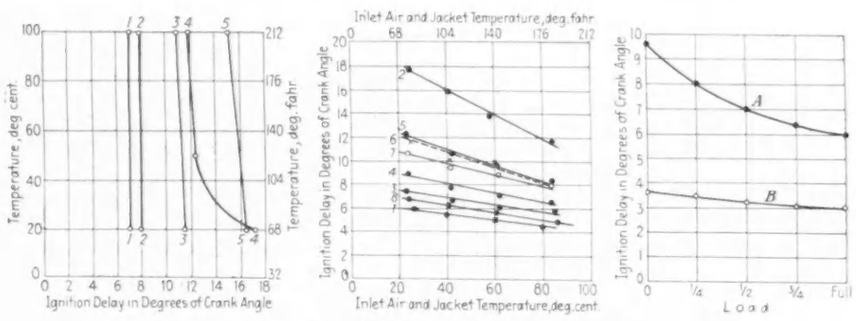


FIG. 4—INFLUENCES OF SEVERAL FACTORS ON DELAY PERIOD

Fuel Temperature.—Fuel 4 Is a Heavy Fuel of 700-Sec. Saybolt Viscosity at 122 Deg. Fahr. Which Apparently Is the Limit at Which Viscosity Influences Ignition with This Injection System. The Test Was Made at One-Quarter Load

Inlet-Air and Jacket Temperatures.—The Test Was Made at One-Quarter Load

Load.—Curve *A* Relates to 25-Cetene Fuel and Curve *B* to 70-Cetene Gas Oil

most simple apparatus for studying ignition. With this engine, injection is largely independent of physical fuel characteristics and can be recorded on an indicator, and the ignition process appeared to be the least complicated. First, a 40-hp. Thomassen engine was used for some years, followed later by a 20-hp. unit, which embodies more variables, such as compression ratio and speed, and which, after a very satisfactory method had been worked out, is now in regular use for the routine testing of fuels. A brief description of the latter engine, from which the first engine differs but slightly, is given with Figs. 2 and 3.

The first fact observed was that the engine would run smoothly on some fuels while on others it ran roughly, and it was ascertained that this smoothness or roughness was only an evidence of a relative fuel characteristic. It was then thought probable that some engine variable could be introduced to influence this fuel behavior and might be used for fuel testing. Main variables for this purpose were injection, fuel temperature, air temperature and air pressure.

Varying the injection appeared to be altogether too complicated.

Varying the fuel temperature proved to have practically no influence. At the left, Fig. 4 shows delay periods for a series of fuels at different fuel temperatures. Save for Fuel 4, having an extremely high viscosity, practically no influence can be detected. It shows the extreme insensitiveness of ignition to viscosity

FIG. 5—DIAGRAMS OBTAINED FROM THROTTLING (LEFT) AND SUPERCHARGING (RIGHT)

within wide limits, visible influence beginning with Fuel 4 at 122 deg. fahr. when the viscosity is 700 sec. Saybolt (20.5 deg. Engler).

Varying the air temperature has a definite effect, as might be expected. At the center, Fig. 4 shows delay periods for a series of fuels at high and at low inlet-air temperatures. At the right are shown delay periods for two rather extreme fuels from idling to full load, which of course is to be considered as another temperature effect. Apart from the fact that the practical limits of air temperature were rather narrow, an important disadvantage of temperature variation for routine tests is the time lag before equilibrium sets in.

Varying the air pressure could be done instantaneously by throttling or supercharging, and produced a very definite change in the behavior of the engine, as was considered desirable for testing the fuels. It must be understood that this variation in pressure implies variations in penetration and other factors, so its effect must be ascribed to complex variations of the process, of which pressure is the simple indicator. Smooth fuels became very rough with throttling, and rough fuels became smooth with supercharging, as shown in Fig. 5.

It was possible to judge exactly at what amount of throttling smoothness ceased, and originally this point was used for rating and was called "knock figure." It was obtained by expressing the critical compression pressure in percentage of the normal compression pressure. Normal fuels showed knock figures between 0.6 and 1.2, the lower figure meaning the better ignition quality in this case. Ordinary pencil indicator diagrams showed the bouncing-pin effect of the sharp combustion on the moving masses of the indicator, which afforded

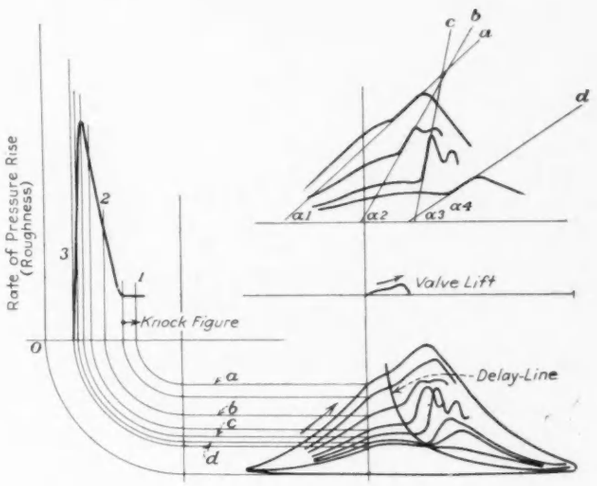


FIG. 6—ROUGHNESS AS A FUNCTION OF THROTTLING

The Roughness Expressed as Rate of Pressure Rise Is Indicated by a Low-Frequency Indicator

a means of expressing roughness in terms of "indicated rate of pressure rise."

Fig. 6 shows how roughness increased with throttling and fell off again. Comparing this with the chart at the right in Fig. 1, it will be seen that, in a throttled test on one fuel, the ignition undergoes a series of stages entirely comparable with such as are due to a series of fuels having decreasing ignition quality in the non-throttled engine.

To obtain more exact diagrams, an optical indicator was constructed with the collaboration of Dr. W. J. D. van Dijck. Fig. 7 shows the principle of this indicator, the pressure-recording element being a membrane with a directly attached mirror. Fig. 8 shows a series of throttled diagrams. Phases 1, 2 and 3 are now clearly seen, and especially the regular shape of Phase 3, insofar as it is present, is now clearly brought out. Incidentally, it is typical that, when fierce knocking occurs,

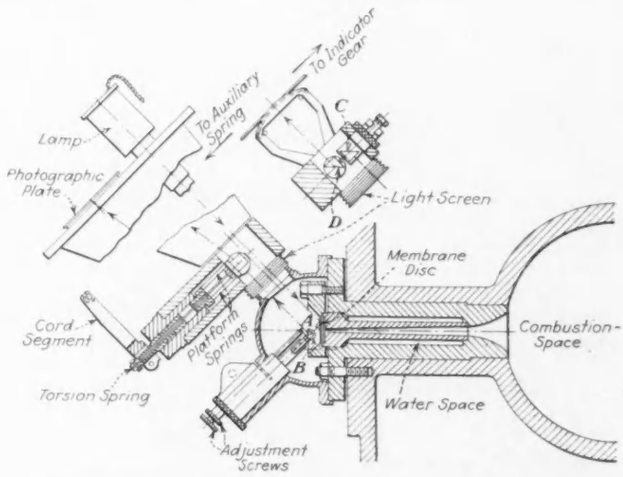


FIG. 7—OPTICAL INDICATOR USED TO OBTAIN MORE EXACT DIAGRAMS

A Beam of Light from a Lamp Is Reflected by Prism *A* Mounted on a Deflecting Diaphragm and Stationary Prisms *B* and *C*. A Fourth Prism, *D*, Mounted on a Spring Platform, Obtains a Reciprocating Movement from a Torsion Spring and Causes Horizontal Deflections of the Light Beam so as To Obtain Closed Diagrams

it appears to be mostly the end of combustion Phase 2 that is responsible for the vibrations and noise of the knock.

Reverting to the diagrams, it is seen that the beginning of ignition, when throttling, follows a steady curve, which we have called a "delay curve" and which has a hyperbolic character. These curves were so regular in character for different fuels that they could be used for rating measurements. This proved that small injection faults due, among other things, to varying amounts of leakage, as shown in Fig. 9, upset the determination of the limit of smooth running (see knock figure in Fig. 6) and that a steady relation existed between delay and roughness, but only under constant-injection conditions. For subsequent work, therefore, delay measurements only were used.

The pencil indicator was found to be perfectly reliable for drawing in the delay line, the faulty indication beginning only after the commencement of rapid pressure rise. This was a really important finding, as, for routine purposes, the photographic indicator could then be discontinued. Throttled diagrams are still used, as we found it easier to draw a delay curve and determine the delay by intersection with a constant-pressure line than to judge where combustion starts on a single diagram. Moreover, the general experience being that delay curves were similar and typical for each fuel, it was possible to measure the slope of the curve, which gave distinct values for every fuel and obviated the indicating of the fuel valve (at the beginning of injection), which was in itself a complication and a source of error. Fig. 10 shows that, on the assumption that delay curves are hyperbolas, measuring the difference in delay between two pressures $2p$ and p gives the same result as measuring delay at pressure p.

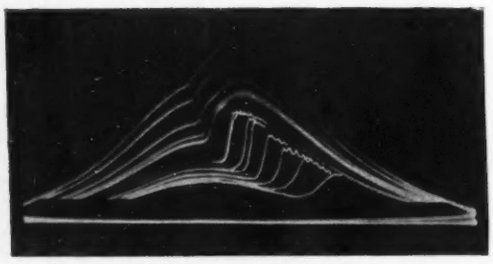


FIG. 8—OPTICAL-INDICATOR DIAGRAM

In a Series of Throttled Diagrams, Increasing Roughness Is Observed. The Phases of Combustion Can Be Observed Clearly

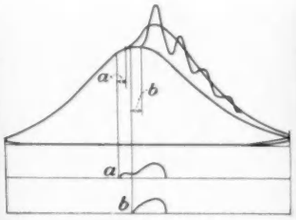


FIG. 9—INFLUENCES ON ROUGHNESS OF FUEL LEAKAGE IN THE HIGH-PRESSURE LINE

Diagram *a* Shows Little Leakage (Viscous Fuel) and Diagram *b* More Leakage (Kerosene)

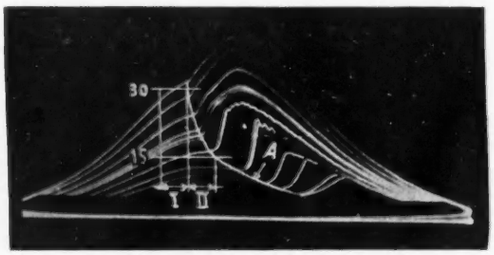


FIG. 10—INDIRECT MEASUREMENT OF DELAY PERIOD

In the Series of Throttled Diagrams *A* Is the Delay Line. The Delay at 30 Atmospheres, Shown at I, Is Practically as Long as the Difference of Delay Periods Measured at 15 and 30 Atmospheres Respectively, Shown at II

According to our experiences, this law has some exceptions, especially with heavy residual fuels, with which the lags at high pressure become too long in relation to the subsequent slope of the delay line, suggesting that the hyperbolas are shifted along the time axis due to the lengthening of the preparatory process of atomization and evaporation. On the new routine engine, pressure and fuel valves are simultaneously indicated on the same diagram; so that, under all conditions of compression ratio, temperature and throttle, true delays can be measured within 0.5 deg.

It was soon found expedient to base all ignition data obtained on reference fuels, for which on theoretical considerations cetene, $C_{16}H_{32}$, and mesitylene, C_9H_{12}, were suggested. This proved to be very effective, as the former substance ignites most readily in a normal engine, the latter not at all.

The routine rating of fuels has now been done for more than a year according to the method of measuring the difference in delay at two pressures, 15 and 30 atmospheres, and expressing the result in terms of reference fuels so as to give each fuel a "cetene number," this being the percentage of cetene in the standard fuel blend which matches the sample tested. The procedure is to run for 8 to 10 min. on each fuel at a low load, taking four sets of throttled diagrams, and subsequently to run a sufficient number of mixtures of standard fuels to make a smooth curve of cetene values for different standard mixtures on a graph. The values obtained from the tested fuels are then plotted on the graph and the cetene numbers deduced, as shown in Fig. 11. For every-day work, cheap substandard fuels have been introduced, for which a conversion curve has been made. In this way about a dozen fuels are tested in a day's work.

Fuel Behavior in Different Engines

As to correlation with different engines and different methods, an ignition test would be useless or at any rate less useful if little or no correlation existed between the behavior of fuels in different engines. In our previous publication[6] we described tests on engines of various types which all point to the existence of a close correlation. We realize that an absolute correlation under every condition imaginable is ruled out completely, but for the moment we think that differences up to at least five cetene numbers are of very little importance. Some typical correlations were the following:

Deutz High-Speed Precombustion-Chamber Engine.—Rated at 20 hp. at 1000 r.p.m. The engine was idling at 1000 r.p.m. and the injection setting was retarded until the engine misfired, which caused blue smoke. This test was rather rough; however, the upper left chart in Fig. 12 shows the various injection settings corresponding with misfiring, plotted against the cetene numbers of the fuels used.

Gardner Hot-Bulb Engine.—Rated at 7 hp. at 600 r.p.m. The engine misfires readily at light loads with many fuels, and an endeavor was made to improve the running by doping such fuels. The upper right chart in Fig. 12 shows the agreement

[6] See *Engineering*, Nov. 13, 1931, p. 603; Dec. 4, 1931, p. 687; and Dec. 18, 1931, p. 755.

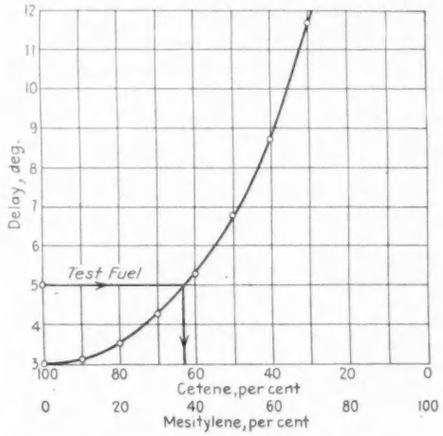


FIG. 11—RATING OF A FUEL IN CETENE NUMBERS

The Result of a Measurement Is Plotted against a Curve Obtained from Reference Mixtures. The Cetene Number Is Then Obtained Graphically

between the quantities of dope and the cetene numbers of the treated fuels.

Thomassen-Ricardo Two-Cylinder Sleeve-Valve Engine.—Rated at 100 hp. at 900 r.p.m. The lower left chart in Fig. 12 shows the agreement between delays measured on this engine and cetene numbers of the fuels tested.

Saurer-Acro Four-Cylinder Engine.—Rated at 55 hp. at 2000 r.p.m. This engine is very sensitive, especially at low loads, when combustion is highly influenced by bad ignition quality. The lower right chart in Fig. 12 shows the fuel consumption of the engine when idling at 1000 and at 1500 r.p.m., increasing with lower-cetene-number fuels.

These and other results showing the various connections between wholly different factors of engine behavior and ignition quality have encouraged us to rate fuels in cetene numbers in a fixed way on the Thomassen engine, where it can be done to a high degree of accuracy. Even if more important discrepancies were to exist in a few cases, it would be better to use one numbered scale which serves as a basis and analyze the anomalies separately than to adopt different scales for different engine types.

Referring to the paper entitled Compression-Ignition Characteristics of Injection-Engine Fuels[7], by A. W. Pope, Jr. and J. A. Murdock, relating to ignition tests in a variable-compression Cooperative Fuel-Research engine, these authors used a promising method of operation and they, too, mention the possibility of using reference fuels; so it seems that substantial agreement

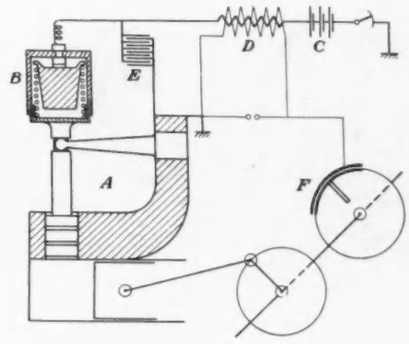


FIG. 13—INERTIA-LAG METER WITH DIRECT NEON-TUBE READING

The Pressure Indicator Is Located at *A*, the Inertia-Lag Meter at *B*, the Battery at *C*, the Coil at *D*, the Condenser at *E* and the Neon-Tube Indicator at *F*

exists about the direction in which a final test method should be sought. On the other hand, we are inclined to think that it would be preferable that their method should be directed toward delay measuring in a regularly running engine and not toward a measurement of a firing limit. The latter measurement probably is more influenced by selective actions of fuel components, like the determination of spontaneous-ignition temperature in heated pots, and there is no certainty that the fuel which ignites for the first time after increasing the compression ratio is forthcoming from the immediately preceding injection, as remaining fuel vapors and half-burnt products may influence results. Moreover, as a practical point for operation, we must warn against the danger of piston gumming due to running at the limit of misfiring.

We are studying the behavior of the Diesel-C.F.R. engine and have constructed a simple instrument which may be called an "inertia-lag meter," shown in Fig. 13. It is fitted in such a way that delays can be measured with the aid of the spark indicator normally mounted on the C.F.R. engine, and consists of a sprung inert mass moved by a pressure indicator, which inert mass breaks a contact at the moment that rapid pressure rise sets in. This occurs independently of the pressure at that moment.

Experiments have shown that the molecular structure of the fuels is the deciding factor as to its ignition quality. Where physical characteristics exert influence,

[7] See S.A.E. JOURNAL, March, 1932, p. 136.

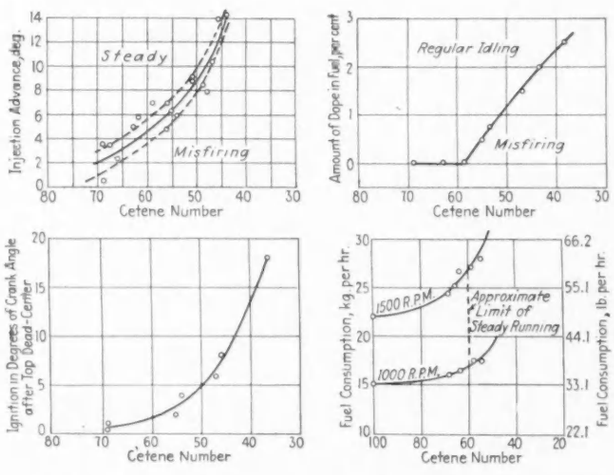


FIG. 12—CORRELATION BETWEEN THE BEHAVIOR OF FUELS IN DIFFERENT ENGINES

Between Deutz High-Speed Engine Smoke Test and Cetene Numbers. — As Shown by the Dotted Lines, 2 Deg. of Crank Angle Are Allowed For Because of the Sloping Cam and Imperfect Observations

Quantities of Dope Necessary in Fuels of Different Cetene Number To Make the Gardner Hot-Bulb Engine Idle Regularly

Beginning of Combustion in the Thomassen-Ricardo Sleeve-Valve Engine versus Cetene Numbers of Fuels

Fuel Consumption of Idling Saurer Engine as a Function of Cetene Numbers

that is, viscosity with air injection, these factors can mostly be taken into account separately.

Our general experience has been that those fuels which will crack readily, or which contain large proportions of readily crackable materials, have the highest ignition qualities, and that those fuels which are thermally more stable will have lower ignition qualities. This thermal stability is dependent upon molecular structure. Straight saturated hydrocarbons are least stable. Double bonds strengthen the molecule. Aromatic rings are still more stable, and likewise very branched molecules. Preliminary cracking of a fuel and removal of the cracked products lower the ignition quality, the result being all the worse according as the cracking has been more intensive.

An interesting experiment was made by mixing increasing proportions of CS_2 in a fuel of high cetene number, CS_2 being extremely prone to oxidation but thermally rather stable. As shown in the chart at the left in Fig. 14, this carbon disulphide has a strongly retarding effect, so that apparently its thermal stability, and not its oxidizing tendency, has been the decisive influence. The relation obtained between cetene numbers of a series of widely differing fuels and the initial cracking velocity obtained from short-time cracking tests at 625 deg. cent. (1157 deg. fahr.) in an apparatus specially built for this purpose by Dr. van Dijck, which proved to be a good means of determining ignition quality by laboratory tests, is shown at the right in Fig. 14. The initial cracking velocity has been expressed in the number of new mols formed, which is a measure for the number of bond ruptures that have occurred.

As a result of these engine and laboratory tests, Dr. van Dijck and the authors venture the theory that, as a whole, ignition in oil engines is the result of a reaction whereby oxygen combines with the free radicals formed by the cracking of the fuel molecules. It is thought that it also may be possible that, in the highly strung state just before its rupture, the hydrocarbon molecule is already abnormally sensitive to oxygen. Anyhow, it is clear that the reaction velocity, and thus the heat formation per unit of time, is proportional to the number of bonds broken per unit of time, so that the cetene number of a blend should be deducible by a linear interpolation from the cetene numbers of the components. In fact, the existence of this linear relationship actually has been found. Various examples will be given farther on. (See upper right chart in Fig. 16).

It is further evident that some fuel should evaporate so as to be cracked at the proper time because, in liquid form, the temperature remains too low. It is interesting to refer to the old theory of P. Rieppel[8] who, in 1908, pointed to the fact that fuels delivering large volumes of oil gases when heated were most suitable for the Diesel engine. Rieppel was severely criticized, but he seems to have been right in principle, although his chemical conclusions were wrongly founded.

[8] See *Meitteilungen über Forschungsarbeiten, Verein Deutscher Ingenieure*, vol. 55, pp. 1 to 35, 1908.

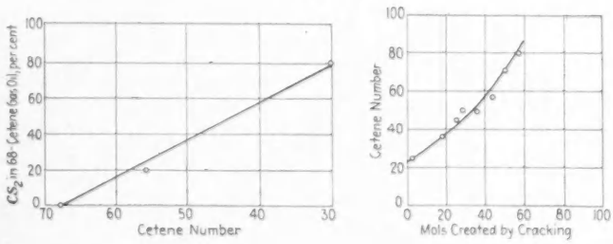


FIG. 14—OTHER INFLUENCES ON THE CETENE NUMBER

Influence of Carbon-Disulphide, CS_2, on the Cetene Number of a Gas Oil

Relation between Cetene Number and Initial Cracking Velocity

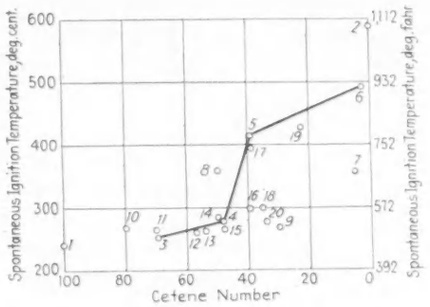


FIG. 15 — SPONTANEOUS-IGNITION TEMPERATURES PLOTTED AGAINST CETENE NUMBERS

The Temperatures Were Determined in a Hot Pot. No. 1 Represents Cetene; No. 2, Mesitylene; Nos. 3 and 6, Substandard Fuels; Nos. 4 and 5, Blends of Nos. 3 and 6; No. 7, Tetra-Iso-Butylene; No. 8, Linseed Oil; No. 9, Carbon-Disulphide, CS_2, Gas-Oil Blend (See Fig. 14, Left View); and Nos. 10 to 20, other Fuels

The foregoing theory may perhaps elucidate discrepancies between measurements of ignition lag in engines or in a special apparatus like that of the Royal Aircraft Establishment (England) as against measurements of the so-called spontaneous-ignition temperature—hereafter abbreviated as S.I.T.—of a drop in a heated pot (See Fig. 15). S.I.T. may be described as the lowest temperature from which slow oxidation will build up enough heat energy somewhere in a given apparatus to reach the initial cracking temperatures. This slow-oxidation heat may depend more on direct oxidizing tendencies of some components than on other factors governing behavior during sudden exposure to high temperature as in the engine (See No. 9 of Fig. 15). At low temperatures and in the very still air of the ignition cup, a selective effect may be expected from easily igniting components which are practically able to retain their own S.I.T., even if present in low concentration. This seems to us to be apparent from Fig. 15, where the jump in S.I.T. between the substandard blends Nos. 4 and 5 of 48 and 40-cetene value, respectively, is remarkable. The quantities of 70-cetene fuel present in those blends are 33 and 45 per cent respectively. Thus fuels Nos. 16, 18 and 20 apparently contain an amount of low S.I.T. fractions which Nos. 17 and 19 do not have; also, volatility influences S.I.T. determinations in open pots. Moreover, the influence of ignition-promoting dopes has often been found to lie more in the lowering of S.I.T. than in diminishing the delay period in an engine.

Investigation of Molecular Properties

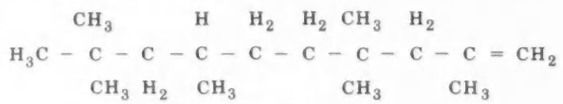
When thermal stability of the molecules is the dominant factor for ignition quality, we must investigate what molecular properties determine this thermal stability. Experiments teach that two properties are of major importance; first, the strength, that is, dissociation energy of the chemical bonds between the atoms, and, second, the spatial grouping of the atoms. This second factor apparently is of even more importance than the first. In general, it may be said that the more compactly the molecule is built up, that is, the more it approaches the form of a sphere, the greater is its thermal stability. On the contrary, the less compact it is, for example, when all the carbon atoms lie in a straight chain, the less is the stability.

The fact being known from optical data in atomic physics that the vibration of the atoms in the molecule can be conceived as the mechanical vibration of masses linked to one another by springs that permit extension and bending, and, further, that the breakdown of a bond is in all probability caused by the synchronizing of the different modes of vibration at one single bond, we can understand that, the more the different modes of vibration are coupled to one another, the smaller the chance will be that all the vibration energy is heaped up in one single bond, causing a breakdown.

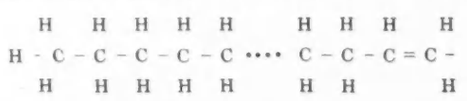
The coupling between the possible vibrations becomes stronger when tertiary or quaternary carbon atoms are present than when only secondary carbon atoms form the links. Thus, the more compactly a molecule is built up, the higher will be the thermal stability and the lower the tendency to crack; that is, the lower the ignition quality.

The nature of the bonds between the atoms, although having a marked influence, is less important. The division in cyclic compounds (such as naphthenes and aromatics) and aliphatic compounds (such as paraffins and olefines), for instance, is altogether inadequate for our purpose. An iso-paraffin or iso-olefine of compact structure may be more stable than a weakly constructed naphthene.

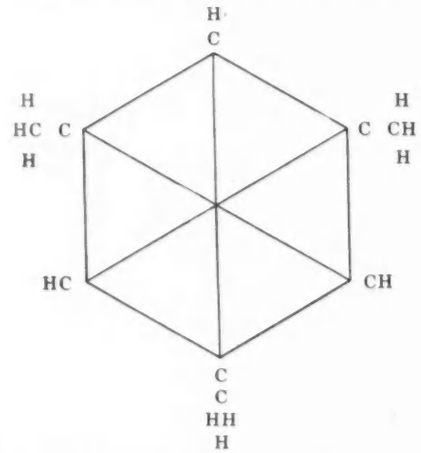
An example is given by a tetra-iso-butylene, $C_{16}H_{32}$,



which happens to have the same formula as cetene



but is of very compact form; whereas cetene has one straight chain. The ignition quality of cetene is the highest yet found by us for any hydrocarbon, that of tetra-iso-butylene being decidedly one of the lowest. Mesitylene, which is used as a reference fuel, is also of a very compact form; the formula being



The influence of tetra-iso-butylene on the cetene number of a gas oil is shown in the upper left chart in Fig. 16. The pure substance could not be used at all. The difference between these two substances at once does away with the notion that the hydrogen content influences the ease of ignition, which was the chemical part of Rieppel's theory and sufficed only to differentiate between gas oil and tar oil. This chart is interesting in that it shows an example of the behavior of blended fuels. These and many other tests were made by our collaborator, J. Ruinen, with great precision. Some other examples are given in the upper right chart in Fig. 16; that is, two conversion curves between standard and substandard fuels and a blending curve of solar oil with Edeleanu extract. It is seen that, practically speaking, the cetene values of blends do not vary from a straight-line relationship, which makes it possible to judge the quality of fuels blended from known components.

TABLE 1—TWO TYPICAL CASES OF BLENDS

Component	Cetene No.	Fuel A, Per Cent	Part Cetene Value	Fuel B, Per Cent	Part Cetene Value
I	40	20	8.0	20.0	8.0
II	55	30	16.5	34.5	19.0
III	62	14	8.7	17.0	10.5
IV	24	14	3.4	14.5	3.5
V	40	22	8.8	13.0	5.2
VI	80	0	0	1.0	0.8
Total (calculated)		100	45.4	100	47.0
Observed Cetene Value			46.0		49.0

Two typical cases of such blends are given in Table 1. It really gives confidence in the figures when one considers the widely varying origin of the substances used

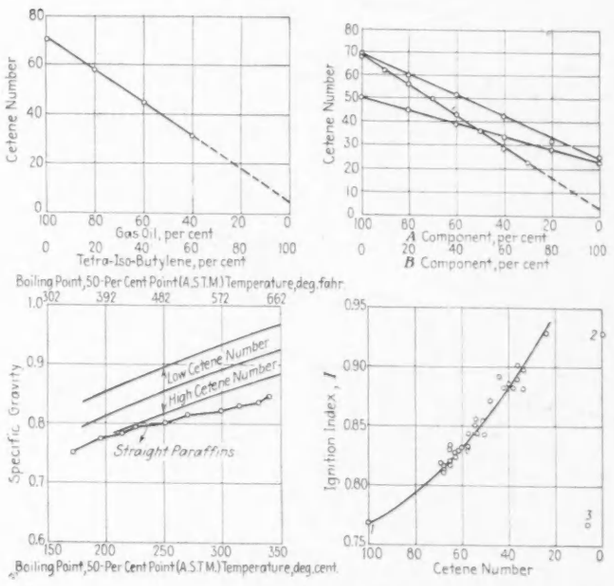


FIG. 16—OTHER FACTORS AFFECTING CETENE NUMBERS

Cetene Number of Tetra-Iso-Butylene-Gas Oil Blends.—The Cetene Number of Tetra-Iso-Butylene Can Be Estimated by Extrapolation

Diagram Showing Specific Gravity Plotted against Boiling Point.—The 50-Per-Cent Point of the A.S.T.M. Distillation Curve Is Taken as the Average Boiling Point for Fuels, so Each Fuel Is Defined by a Point. The Points for Fuels with Equal Cetene Numbers Are Found within Small Limits from the Drawn Lines

Straight-Line Relationship of Fuel-Blend Cetene Numbers

Rough Relationship between Cetene Numbers and Ignition Index, I, for a Number of Fuels. — The Index I Is the Specific Gravity Corrected for Average Boiling Temperature; That Is, $I = G - 0.0075 \times [50\text{-per cent A.S.T.M.} - 250 \text{ Deg. Cent. (482 Deg. Fahr.)}]$. The Unreliable Nature of This Relationship for Hydrocarbon Fuels Is Apparent from the Values Indicated at Points 2 (Mesitylene) and 3 (Tetra-Iso-Butylene) Which Are Widely Out of Line. Point 1 Represents Cetene

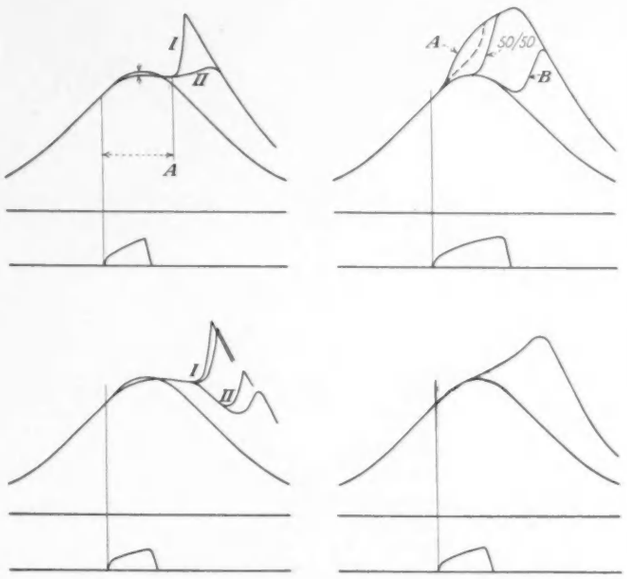


FIG. 17—DIAGRAMS RELATING TO DELAY PERIOD AND IGNITION

Delay-Period Chart Showing Pressure Drop Due to the Cooling Action of Fuel, Subsequent Slow Pressure Rise and Important Change in Reaction Velocity at *A*, in Process I, with Rapid Flame Development. A Process Like That Indicated by II Should Be Expected if Every Part of the Fuel Charge Were Left To Provide Its Own Ignition, Interheating Action Being Supposed To Be Excluded

Diagram Showing the Influence of Blending on Delay. —A 50-50 Blend of Fuels *A* and *B* Does Not Start at the Same Delay Period as Fuel *A*, but Starts Later

Extreme Type of Reaction.—In this Type the Flame Would Be Gradually Built Up in the Entire Combustion Chamber

Irregular Ignition. — This Is Due to the Irregular Spray, Showing Sometimes Nuclear Ignition at I, Sometimes Failing of the First Nucleus, and Subsequent Ignition at II from a Less Advantageous Nucleus

for these tests, among these being cetene derived from spermaceti, coal-tar oil and mesitylene derived from coal, three gas oils from Venezuelan and Eastern crudes and an Edeleanu extract.

Fuel Formula

For straight distillate fuel oils from natural crudes, a simple formula may often serve for estimating the ignition quality.' On plotting the specific gravity against the 50-per-cent A.S.T.M. point *T*, it was found that a sheaf of lines could be drawn through fuels of equal cetene number, as shown at the lower left of Fig. 16. These lines are of the type

$$G - aT + bT^2 = \text{a constant},$$

where *G* represents specific gravity.

For gas oils the term bT^2 is of no importance, and for a large part of the straight lines

$$G - 0.0075\ T = \text{a constant}$$

may serve as well, whereby a high constant means a low cetene value. This led us at the time to adopt an "igniton index", *I*, the formula being

$I = G - 0.0075$ [50 per cent A.S.T.M. − 250 deg. cent. (482 deg. fahr.)]

The constant subtraction of 250 deg. cent. was introduced merely for the sake of convenience.

The relationship given in the lower right chart in Fig. 16 between index *I* and cetene numbers shows a happy coincidence for natural hydrocarbon fuels. However, the application of the formula has its limits; naphthenic fuels are rated too high and mixtures of straight paraffins and aromatics too low. Cracked products and pure substances of a typical character, for example, iso-paraffins, may fall out altogether. Tetra-iso-butylene is seen in the chart as having the same index *I* as cetene, which is its antithesis as regards ignition quality, as may be evident from the foregoing theory. For distillate fuels the average values obtained agree generally within some 10 cetene numbers. As a whole, however, it is not safe to make use of such formulas.

Ignition

The particles (free radicals) of a molecule cracked by heat have at the moment of being formed, according to our conception, a reactivity of a very different order from that of an unbroken molecule. As soon as fuel is injected, part of it rapidly attains a temperature high enough to initiate cracking. The vapor that is formed, partly by the very action of atomizing and partly due to the movement of the drops through the hot air, is in this respect in a favorable condition. The liquid drops themselves can be considered to remain quite cold from the viewpoint of cracking. They are evaporated rapidly enough after the flame has formed. During the delay period the quantities of heat given off by the beginning of combustion serve to increase the temperature on the spot and will promote evaporation and cracking. Apparently, some definite temperature must be reached somewhere to change suddenly the velocity of reaction from the low velocity during the delay period to the high velocity during Phase 2, as indicated in the upper left diagram in Fig. 17.

Nuclear Ignition

It is conceivable that, after sufficient temperature has been reached in some spots with the required mixture strength, these spots build a nuclear flame by which the other parts of the fuel are ignited far more rapidly than they could ignite themselves. If this interaction did not take place, the diagram to be expected would be rather more like process I in the upper left diagram in Fig. 17 where each portion of the fuel ignites itself, with delays that gradually shorten owing to rise in average temperature. This happens only after very long delays, which appears to show that the normal case is that of nuclear ignition.

There is some typical analogy with the spark-ignition engine in that, in compression-ignition engines also, combustion may often be considered to need the formation of such a nucleus, and Ricardo has pointed out this fact in his work. When such a flame is formed in a Diesel engine it usually causes such rapid burning that, after practically all the fuel present has burned, the rest of combustion follows injection closely as long as there is air enough. Two diagrams are reproduced at the upper right in Fig. 17, fuel *A* igniting early and fuel *B* extremely late. As pointed out for a blend, the reaction velocity, that is, the heat generated per unit of time, can be calculated from the reaction velocity of the components by linear interpolation. Therefore, for a mixture, the lag, which is inversely proportional to the reaction velocity, should lie between the lags of the components and it is not correct to assume that the ignition is started by the fuel having the shortest delay. It is not the dotted line but the intermediate full line which expresses the behavior of a blend in practice.

Sometimes, when a fuel spray is irregular and some parts are more or less separated from the main body, this phenomenon can be observed in another way. In the lower left diagram in Fig. 17, a particularly favorable nuclear flame is formed which ignites the whole charge at point I. We get the impression that, if this fails, the nuclear flame is apparently burnt out and the next best must take up the work, which it normally does at point II, with spreading results.

It is again conceivable that this is not the only process possible and that, with extreme conditions such as very high turbulence, hot combustion-chamber walls and the like, the phenomena described are liable to change in magnitude or character. Diagrams of engines with extremely high turbulence would seem to approach more the form depicted in the central chart in Fig. 1, which might mean a growing of the nuclear flame in dimensions due to combined evaporation-increasing and temperature-leveling-off effects of turbulence. This promotes reaction only as long as it does not succeed in building a flame, and finally would approach the form depicted in the diagram at the lower right in Fig. 17. This would be the extreme type of a reaction, so leveled-off over the whole of the combustion chamber that it accelerates with the rise of the average temperature and loses its typical character of combustion after nuclear ignition. This may, in our opinion, be partly the case with the beautiful constant-pressure diagrams of air-blast Diesel engines.

Special Cases of Ignition

Some curious ignition phenomena with regard to ignition problems may be mentioned. The first case is that of the influence of viscosity on the ignition of an air-blast-injection Diesel engine. Fig. 18 shows diagrams obtained when running on a gas oil and on a heavy fuel. The delay period of the gas oil is very small; that of the heavy fuel long, with a heavy knock afterward. Injection occurs after ejector action in the atomizer, a differential pressure causing fuel to enter through a gap into the blast airstream. Obviously, with a constant gap, this causes a more viscous fuel to be injected more slowly, which means that it is not able to attain in time the temperature of a nuclear flame, which moreover is influenced in the wrong direction by the cool injection air. Our collaborator, B. J. J. Gravesteyn, constructed a variable-gap injector with which the gap can be adjusted while the engine is running. On adjusting the gap to a width corresponding to the viscosity of the fuel, the delay is brought back to that of the gas oil. As the viscosity and most probably the fineness of atomization have not been changed, the important influence appears to have been the speed of injection.

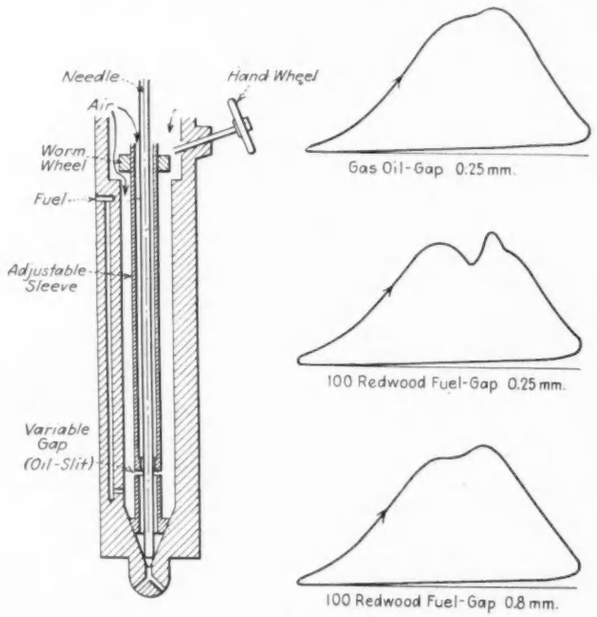


FIG. 18—ADJUSTMENT OF AIR-INJECTION ENGINE TO VISCOSITY

A Section of the Variable-Gap Injector and Diagrams Taken on the Werkspoor Engine Are Shown

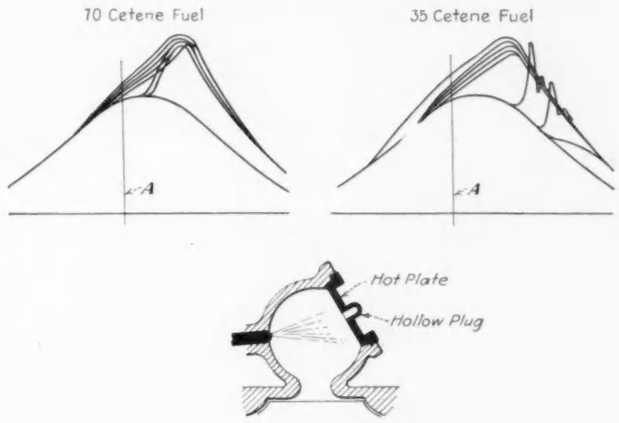


FIG. 19—IGNITION BY FLAME FROM FUEL REMNANTS USED IN THE KROMHOUT SEMI-DIESEL ENGINE

Two Diagrams on Widely Different Fuels Show Insensitiveness Obtained by Flame Ignition before Injection

Hot-bulb engines show curious ignition phenomena. They are known to be unstable when operating with a very early injection setting, because an early-igniting fuel will burn before top dead-center and heat the bulb more than will a late-ignition fuel, thereby accentuating the difference between the fuels. Traces of this instability can be found in Diesel engines at part load by measuring the heat lost through the combustion-chamber walls. With a late-igniting fuel, fewer calories will be abducted by the cooling water than with an early-igniting fuel.

With some fuels containing ashes of certain kinds, troubles have been experienced in hot-bulb engines because these ashes form crusts on the hot bulb. In some cases this caused preignition through glowing; in others, misfiring through insulating the spray from the hot walls.

Some types of hot-bulb engine function on an entirely different kind of ignition, caused apparently by a fuel remnant deposited in a niche during the injection. This fuel finds insufficient air in this niche, extinguishes and flames up again when fresh air is introduced during the next compression stroke.

Fig. 19 shows two diagrams obtained on our Kromhout semi-Diesel engine, on a 70 and a 35-cetene fuel respectively. It will be seen that ignition starts normally before injection and that it is only when this pilot ignition flame fails that the low-cetene fuel betrays itself by knocking. In extreme cases one can imagine that a certain engine will run smoothly on a low-cetene fuel in the way described; whereas it will show rough running on a fuel of higher cetene value, which would burn away too early from the niche to be able to ignite the fresh charge. This ignition before injection possibly explains why the hot bulb should be brought to such a high temperature when starting; whereas, after-

ward, a much lower temperature will assure regular ignition.

Cleanliness of the nozzle is a prime essential to obtain fair ignition values for different fuels. It is often observed that, as the nozzle becomes carbonized, knocking becomes irregular and extremely early ignitions occur, alternating with the normal ones. This leads to the supposition that the nozzle deposits are actually burning or smoldering, as indicated in Fig. 20, and sometimes ignite the charge. On the other hand, it has been observed that carbonization on the plain nozzles of an antechamber engine caused extremely rough running due to late ignition after a defective spray, so that there is no definite sense of the influence of nozzle carbon on ignition. In all cases, however, performance of the engine is affected by deflection of the spray and incorrect mixing.

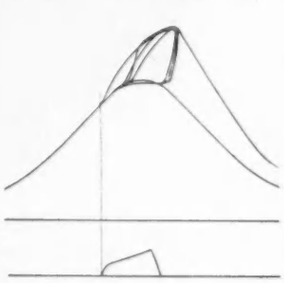


FIG. 20—IRREGULAR EARLY IGNITION IN A SOLID-INJECTION ENGINE

This Is Caused by a Carbonized Nozzle, Which Probably Retains a Smoldering Flame. This Type of Irregularity Is Entirely Distinct from That Shown in the Lower Left Chart in Fig. 17

As to the relative importance of ignition quality, we have attempted in the foregoing text to give a short review of our researches on ignition in oil engines. It should be emphasized here that, although delay influences combustion, especially in high-speed engines, and excessive delay leads to incomplete combustion, smoke, odors and gummed pistons and valves, yet the ignition quality is not all-important. Often a certain fuel is found to give better all-round results, including nozzle troubles, cylinder wear and complete combustion at high power, than another of higher ignition quality. This should be yet another argument for trying to solve the delay problem from the engine side so as to be able to use the products of best general quality. No gain in efficiency or power is to be expected from higher-ignition-quality fuels. Therefore, if a tendency to ask for special-ignition-quality fuels should arise, the engine that has a wider range in this respect of ignition quality would immediately score.

General Remarks on Diesel-Engine Design

In trying to draw some conclusions of interest to designers of solid-injection engines, we are obliged to make use now and then of views published by others. First, let us review what phenomena are presumed to take place within the engine, then what desiderata are to be considered, and finally try to combine those two groups so as to obtain guiding principles for designing.

As regards the phenomena, the following may be briefly stated: Ignition and combustion differ in character and requirements, ignition being dependent principally upon the chemical structure of the fuel, air temperature and pressures, and combustion on correct mixture strength and turbulence. Knock is caused by delay during ignition and is the effect of a too-rapid change in pressure rise. In combination with a too weakly constructed engine, a too-rapid pressure rise may lead to what is known as rough running. After a certain delay the magnitude of knock depends on the rapidity of combustion, which is governed by mixture strength and turbulence. In fact, it is not yet known whether detonation in Diesel engines has the same explosive character as in gasoline engines; but, practically, the name of detonation may be given for the moment to the extreme form of a too-rapid change in pressure rise, the sound often being equal in the two cases. Bad combustion may be due to faulty ignition or to bad mixing.

The top of the spray is partly gasified and, where spots of the required mixture strength reach a sufficiently high temperature, a nuclear flame will be born. Whether this flame is built up on minute drops or from gas is in this respect immaterial, although, as stated before, we are inclined to accept the latter. Generally speaking, the finer the spray is, the easier is the ignition.

It is important to distinguish between several different types of movement of the air in the combustion space; they may at least be classified as follows:

(1) Stagnant air
(2) Local turbulence of relatively small radius
(3) Local turbulence of relatively large radius
(4) Streaming turbulence through passages
(5) Slow regular swirl

Stagnant air in the combustion space generally shows good ignition but not the highest efficiency of combustion, due to defective mixing and lack of flame velocity.

Local turbulence with relatively small radius will promote the relative movement between the fuel and the air and thereby gasification; it will promote ignition by rapid formation of a nuclear flame and accelerate combustion.

Local turbulence with relatively large radius, on the contrary, even when aiding evaporation, may yet disturb the formation of an effective nuclear flame, thereby causing delay and rough combustion.

Streaming turbulence through passages will exchange much heat with the walls and, particularly when gases are at flame temperature on the expansion stroke, will heat the walls to a high degree. It may be of use for the heating of an "ignition corner" or for bringing specially hot air into an ignition corner, or for atomizing, mixing and burning unburnt fuel generally.

A regular uniflow swirl is rather inactive as to heat exchange with the walls, but is very useful as a means for mixing half-burnt combustibles with fresh hot air for complete combustion. Thus, different kinds of air movement and turbulence may be differentiated as to their effects on mixing, heat transfer to and from walls and flame propagation. They must also be analyzed as to their origin and may be classified into intake, compression, injection, combustion and expansion turbulences. Of these, injection turbulence should be mentioned specially, as, unlike the other kinds, it is not known from gasoline-engine practice. In air-injection engines its presence is evident, but also in solid-injection engines, owing to the rapid movement of small globules through the air, there should be a fairly high amount of the small-radius turbulence of the second kind, already mentioned.

Turbulences caused by compression, combustion and expansion play a more important rôle than in gasoline engines, especially when antechambers or separate chambers with throats or passages are used.

Generally speaking, too strong an air movement may lower the temperature in the surroundings of the birth of the nuclear flame and spoil ignition conditions; it will also roughen combustion for a certain delay. Too slow a movement may spoil combustion by insufficient mixing. At the end of the injection the particles last projected, if not caught in an atmosphere sufficiently rich in oxygen to prevent escape, will cause incomplete combustion and smoke.

Besides compression temperature and pressure, wall temperature should be considered as an important factor, as is evident from the fact that antechamber engines are bad cold starters. Distribution of wall temperature is to be classified as in gasoline engines, with the highest temperatures for the exhaust valve and the

piston center and the lowest for the intake valve. In addition, we may find in Diesel engines walls of artificially high temperature, as, for example, parts of precombustion chambers and walls of passages. Air contact with the walls will result in temperature differences in the air charge. Local mixing with residual exhaust gas will increase the temperature locally and thus, under favorable circumstances, promote surer ignition, although it probably will cause a decrease of combustion speed because of diluents. The designer should apply these different types of turbulence and different wall temperatures consciously as constructive elements.

Combustion-Space Processes

As to the processes taking place in the combustion space, the following main types may be distinguished:

(1) The precombustion-chamber type with narrow or wide passages, the chamber containing only a small part of the compressed air; injection is in the chamber.

(2) The separate-chamber type with one wide passage, the chamber containing practically all the compressed air; injection is in the chamber.

(3) The Acro type with one wide passage, the chamber containing practically all the compressed air; injection is in the passage.

(4) The Verdränger type with the valves in the chamber and with a streaming turbulence produced by a special projection on the piston crown, fitting into the throat of the combustion chamber.

(5) The pure direct-injection type without a uniflow swirl.

(6) The pure direct-injection type with a uniflow swirl.

It would be interesting to analyze different engine types as to the use the designer has made consciously or unconsciously of the different air movements and of the distribution of heat and hot walls in the combustion chamber. This, however, is beyond the scope of this paper. Moreover, a very careful study of air velocities and temperatures would be necessary for a thorough analysis.

When considering what may seem desirable, we want first to point out that too rapid a change in pressure is to be avoided, yet at the same time the maximum allowable rate should be attained. Thus, changes in rate of increase and decrease and of pressure are required up to the limit of rough running. Therefore it is necessary to govern ignition speed as well as combustion speed, as far as possible by engine characteristics, practically in the same way as is achieved for valve movement by effective cam profile. Governing ignition speed means governing delay.

A very fine spray is desirable for a very short delay; but, from a practical viewpoint, extremely fine holes in a nozzle are not desirable. For this reason, in small engines, a single-hole nozzle is to be preferred to a multi-hole nozzle because of its self-cleansing properties. Actually, some other method of attaining short delay should be preferred. The use of fuels with special ignition quality is not essential, as it does not raise the thermal efficiency of the Diesel engine; it is not desirable from a viewpoint of cheap fuel supply and is not necessary from a constructional standpoint, as certain high-speed Diesel engines already exist which will consume without difficulty fuels with almost any ignition quality.

Therefore, for shortening the delay, the use of high-temperature surroundings for the birth of the nuclear flame seems to be dictated. Moreover, perfect combustion should be assured by perfect mixing, which demands effective air swirl after ignition.

Cold starting is extremely desirable.

Injection should be directed so that, even with slightly imperfect spray, unburnt fuel cannot reach the cylinder walls and cause dilution, smoke, carbonized piston-rings and other troubles.

Guiding Principles Stated

To develop some guiding principles, let us now try to combine the phenomena that we presume to take place with what we consider desirable.

In gasoline engines, one of the main objects of combustion-chamber design is to prevent the unburnt part of the charge from detonating. Therefore this chamber should be formed so as to protect this part of the charge as far as possible against a rapid rise in temperature. Considerations in regard to swirls, ratio of cooling surfaces to volumes, progress and radiation of flame and the like, point to the necessity for lengthening to the extreme the delay of detonation to avoid knocking.

In Diesel engines, on the contrary, one of the main objects of combustion-chamber design should be to shorten the delay to the extreme to avoid knocking.

As practical limits to the fineness of a spray exist, every effort should be made from a constructional viewpoint to promote a high temperature at ignition. Therefore the aforementioned special restrictions enforced in designing the combustion chamber of gasoline engines do not apply with Diesel engines, so their combustion space proper might have the most appropriate form for effective charge and discharge and complete combustion. However, other restrictions are to be observed. Since high-temperature surroundings are required for the birth of the nuclear flame, the aim should be to build together such parts of the combustion chamber as are predestined for this purpose by their exceptionally high temperatures. This part of the combustion-space might be called the "*ignition corner*". In addition to natural hot walls surrounding the ignition corner, such as exhaust valve and piston center, in Diesel engines, artificial hot walls may be used; for example, precombustion-chamber walls, sleeves, passages, ridges and the like.

It is self-evident that the intake of the cold-air charge should be such as to cool the ignition corner as little as possible. Further, since for complete combustion an effective swirl is required, the aim should be to construct in such a way as to have a good swirl, but to prevent this swirl from disturbing the ignition corner before and during ignition.

Cold starting points to the direct-injection types of engine.

Protection of the cylinders against defective spray points to the precombustion and separate-chamber types of engine; also, a rim on the piston may help to protect the cylinder wall.

To combine all these conflicting elements is impossible without compromising and, for different cases, different designs may be preferred. Thus, for engines running mostly under full load, the piston top may be a useful hot part of the ignition corner, while, for engines often running under low load, it may, on the contrary, be too cold for this purpose and constitute a source of trouble. Direct-injection engines probably will give highest power and lowest consumption; on the other hand, engines with a separate chamber may be more foolproof and show a greater flexibility. An important field for designers to explore still remains, and application of the principles discussed herein may lead to still higher efficiency than has so far been obtained, with practical insensitiveness to the ignition quality of fuel.

Bodies Considered from Car Buyer's Viewpoint

By J. W. Frazer[1]

Detroit Section Paper

BODIES that are simplest in outward appearance and particularly in interior details are not always the most popular with our American public. The designer, if his product is to appeal to the great masses, is faced with the necessity of striking a happy medium between the ornate, which must not be accused of being gaudy, and the simple, which cannot go so far as to be in good taste.

From my contact with the public and with dealer organizations, my opinion always has been that the responsibility of the sales manager of an automobile company is not limited to the acceptance of a product as it is designed and manufactured and then trying to convince the public that it is what it wants, but that he should make clear to the creators of our designs what he thinks the buying public wants, so that as much sales resistance as possible will be removed.

Evolution Should Be Gradual

First, the general appearance must please buyers of two types; the first are the men who expect to buy your car this season; second, the buyers who have bought your last year's car. For the former you must have a design that is different from the present model, but it must not be so different that it will make your last year's car obsolete, thereby losing the friendship of the buyer of yesterday and possibly his business of tomorrow. So, if we admit, as I think most car buyers will, that the air-flow or tear-drop design is to be the design of the future, our evolution to this end should be by slow steps from the imitation horse-drawn vehicles that we have been building in the past. This will have to be done gradually; we shall have to retain on each new model something we have had on our last previous model, yet our progress should be definitely toward the goal we seek. For instance, the sloping windshields and the curved back of the sedan bodies of today are acceptable to our public and very definite steps in the direction of reducing wind resistance.

Builders of smaller-production cars can forge ahead more rapidly than the large-production designers dare to do. As every large market includes a considerable number of individuals who will be interested in something radically different from their neighbors' possessions, the small produer is able to make radical changes, but in his endeavor to be different in design he runs a great risk of producing something that will be accused of being bizarre; and if he gets this reputation he will find that, although the car was one of the notable attractions at the automobile shows, his sales will dwindle as the season progresses. Danger also exists that the first purchasers of his product may tire of its appearance and will not induce their friends to buy.

Buyers Like Impressive Details

One would think that, with the great pressure brought by the public upon automobile engineers to develop ease of control and handling of an automobile, a demand would also arise for the elimination of gages and indicators which go in the instrument panels, and that, if these gages are necessary, they should be as inconspicuous as possible. However, such is not the case; vanity comes into play, and almost every man or woman who drives an automobile likes to see an impressive row of gages and dials in front of him or her, probably because it suggests great power and speed under expert control. An interesting fact in connection with the sale of higher-priced cars is that any man who has had any experience with a yacht wants an engine-turn instrument in the panel. The best policy is not to design the panel to look too simple. Sometimes I feel that we would do better if we would add another gage or two.

Next in importance to the instrument panel is the steering-wheel, which, to the owner, is the most personal part of the automobile; and nowadays most car owners are drivers. Attention can be given with advantage by the designers to the spark and throttle levers and to the size and also the spokes of the steering-wheel. At a certain point in the making of every sale, whether in the showroom or on the road, a good salesman places the prospect in the driver's seat and wants him to clutch the steering-wheel. If the effect is pleasing, the sale usually is made. Designers heretofore have not protected their salesmen in this particular respect as they should have done; steering-wheels have been too cheap and not enough thought has been given to their design.

Preferred Upholstery and Fittings

Today's prospective buyers want a bright interior. Light-colored upholstery materials are in demand. My observations show a very definite trend from plushes to flat cloths or cords. Those customers in certain parts of the United States who prefer plushes want a long, silky pile—not a short, stubby pile—regardless of how much better the latter would wear. Above everything else, the purchaser of today wants a robe rail that is securely fastened; not one that can be pulled off after the first month or two of use. We are selling now to a sophisticated public that has used automobiles before, and customers know the points of weakness in the old cars and those that are most important in the new cars.

Dual-position foot-rests always appeal to the feminine buyers, but vanity cases are almost out of demand at present.

Ash trays on both sides are desirable, and cigar-lighters often have considerable to do with finally closing a sale. For the salesman to say that a cigar-lighter can be put in the car is not sufficient, as a purchaser has the impression that if these accessories are installed at the factory the fixture is much better and more permanent than if it is attended to at the dealer's place of business.

Carpeted front compartments have closed many a sale because women are more impressed by a detail of this kind than by transmissions, rear axles or other mechanical features that cost much more. While the salesman is telling her about the horsepower or the transmission, she is wondering how she could operate the accelerator without ruining the heel of her white slipper.

The usual owner, after taking delivery of his car, very seldom changes the position of his front seat, but any adjustable front seat that the prospect can operate on the salesroom floor is an excellent psychological aid in closing a sale.

I want to sound a salesman's warning to designers against putting knick-knacks in automobile interiors. Even buyers in the low-price class have become some-

(*Concluded on p. 299*)

[1] A.S.A.E.—General sales manager, Chrysler Sales Corp., Detroit.

The Rational Specification of Airplane Load Factors

Discussion of Edward P. Warner's Aeronautic Meeting Paper[1]

CHAIRMAN T. P. WRIGHT[2]:—For me to express adequately our deep appreciation to Mr. Warner for his remarkable paper and the talk he has given us is obviously impossible. Whenever anyone deals with fundamentals as adequately as he has done, he certainly is entitled to our respect and thanks, particularly when we know of the great amount of work in all branches of aviation that Mr. Warner is performing.

Analyses of airplane failures in the air bring to light the fact that between 4 and 9 per cent of all accidents can be attributed to structural failures. This is, perhaps some might think, a relatively small percentage, but it is an extremely important percentage because probably no type of failure is so fearful to the average passenger as the thought of part of the structure giving way. Engine failure or pilot failure, and many other classes of failure, may be far more important in percentage, but I feel that they do not have the real importance that structural failures have in the minds of the laymen; therefore anything we can do to lessen this small percentage is obviously essential to have carried out.

Our structural analyses and the various rules governing them are, fortunately, brought to a very satisfactory state of affairs through the leadership of such men as Niles and Newell, but nevertheless the decided importance of eliminating structural failures makes it absolutely essential that we be ever vigilant in determining new means of correcting our present methods of analysis and of instituting researches which will bring to light new forces and new methods of analysis.

The classes of faults that may be found in a structure can be divided into three groups. The first of these embraces those pertaining to workmanship and material. Obviously, the remedy there is inspection. The second class includes faulty analysis assumptions which lead to faulty load factors, or inadequate or even absent load factors. Mr. Warner's paper deals with this class. The third class has to do with incorrect design.

Meeting a Gust in a Dive

An airplane has to operate in three dimensions. Obviously, therefore, a great many more possibilities of its getting into situations that will cause different types of loading on the structure are presented than obtain for structures of the older branches of engineering, such as bridge work or buildings; hence a study of these conditions is of fundamental importance in the whole matter of structural analyses.

One point in the paper which I noted as being perhaps one on which Mr. Warner could give some more light is the possibility of combining conditions 1 and 4; that is, the condition presented when the plane is in a vertical dive and meets a horizontal gust. It is a condition that those who are engaged in the construction of naval planes have recently had decidedly brought to their attention; namely, the application of a horizontal gust of 30 ft. per sec. when the plane is in vertical dive, which in many cases becomes a designing condition.

EDWARD P. WARNER:—As Mr. Wright has said, we have had a recent reminder that this condition can exist, and certainly we should take it into account in analysis. I have no excuse to offer for having left it out, except that I get back again to the gloomy conclusion that we have to stop somewhere; just as we cannot take account of what a pilot is likely to do if he dives out of a cloud in a commercial plane and suddenly sees the ground about 200 ft. below him, so we cannot take account of every atmospheric condition. The hazards of military and naval flying always will be peculiar. I think we shall have to take into account only to a very limited extent the possibility of striking a very severe horizontal gust while doing a certain vertical diving maneuver, or we shall get structures so heavy that their loss in military efficiency will more than counterbalance any gain in structural safety. I think we should have some study of that characteristic, but my guess is that, when we have analyzed it, we shall find that we are limited to taking as our basic condition a gust of perhaps 10 or 15 ft. per sec.

Would Specify on Possible Elevator Movement Rather than Stick Force

JOSEPH S. NEWELL[3]:—Mr. Warner's three stipulations as to the limitations of diving or gliding speed for airplanes of different categories are excellent and must be considered in the final analysis and disposition of the problem of rationalizing load factors. That even passenger transports should be able to dive through a hole in fog and be pulled out after a dive of 3 or 4 seç. seems reasonable and should be considered as a possible minimum requirement for such aircraft. Such a dive has been mentioned in previous discussions of load factors and has generally been rejected as being impossible of execution with passengers. More recent experiences seem to justify its inclusion in a rational set of requirements, and the possibility of encountering bumps due to inclined or horizontal air currents during such a dive should also be considered.

It is now generally admitted that the pilot's physical limitations will establish the upper limits for load factors on airplanes of all types. For small, highly maneuverable types his ability to withstand high accelerations will be the controlling factor, while for the larger, more unwieldy planes the pilot's ability to exert a force on the control system would govern. The latter factor, is, however, difficult to establish and, while Mr. Warner's suggested value of 100 lb. is reasonable, the fact should be pointed out that a stick force of 100 lb. means widely different hinge moments at the controls. The use of balances of any type or the probable incorporation of a servo-mechanical device on very large aircraft would completely vitiate the effect of a design specification based on a limited stick force. It seems more rational

[1] Published in the S.A.E. JOURNAL, April, 1932, p. 171. The author, a Member of the Society, is editor of *Aviation*, New York City.

[2] M.S.A.E.—Vice-president and chief engineer, Curtiss Aeroplane & Motor Co., Buffalo.

[3] Associate professor of aeronautical structures, Massachusetts Institute of Technology, Cambridge, Mass.

to base the specification on possible elevator movement and to assume that the full movement available to the pilot is possible, particularly in an emergency such as diving through a hole in fog and finding the ground much closer than was anticipated. Experience has shown that pilots, when frightened by jammed control wires or the sudden appearance of an obstacle in a fog, can exert a force of 200 lb. or more on the stick, and cases are on record in which the pedestal holding the stick has been pulled away from its supports in such emergencies.

The physical limitations of the pilot vary widely in such cases, depending on his mental attitude; so it is believed to be better to base requirements which are to depend on the controlability of the airplane on its absolute limits so that artificial stimulation of the pilot by balance controls, mechanical devices or plain fear will not nullify the point desired in the specification.

The conditions encountered in bumpy weather must be given further attention, and Mr. Warner's suggestions with regard to this are excellent. Present knowledge of the subject, based on data accumulated during the last two or three years, indicates that flight through disturbed air probably will furnish the criteria for the minimum requirements for both upward and downward forces acting on an airplane in normal flight. Further study of this problem, including the procurement of the required meteorological data, is necessary and it is hoped that it will be carried forward as rapidly as possible.

Factor of Safety Varies with Materials

As for factors of safety, too little has been said in the past. The author's suggested values of 1.25 for recovery from a dive and 1.5 for bumpy air seem reasonable for the conventional airplane having a primary structure of metal or wood with a fabric or other non-stress-carrying cover. However, for the monocoque, or stiffened stress-carrying skin, the problem is more difficult and depends on whether ultimate stresses or elastic limits are to be used, whether a smooth skin is to be permitted to wrinkle, so long as the wrinkles do not become permanent, or whether no wrinkling is to be assumed. With the wide range of ratios of ultimate stress to elastic limit for the different types of stress-carrying covering, the incorporation of any safety factor in the load factor and the use of ultimate stresses as allowable stresses become difficult to justify because of their inconsistency. With some materials and heat-treatments, the ratio of ultimate tensile or shear stress to elastic limit is 1.5, whereas for others it is about 1.2, which is a range too great to be ignored and one of extreme importance when determining load factors, since the integrity of the structure depends as much on the magnitudes of the allowable stresses adopted as on the determination of the magnitudes of the design loads. Mr. Warner discusses the need for considering this matter, and I desire to emphasize the absolute necessity for doing so before rational factors can be evolved.

We are now at a fork in the road, one path leading to the development of airplanes of the types that have been conventionalized since the war, and the other leading eventually to the successful development of the newer, stressed-skin, all-metal types. The latter, when designed by methods evolved from successful practice with fabric-covered aircraft, are frequently heavy and generally lacking in uniformity of strength throughout their structural elements. To improve the standard of structural design for such types, we must improve our knowledge of both loads and materials. Mr. Warner's paper contributes a number of excellent suggestions adding considerable to our knowledge concerning the upper and lower limits of the loads for which airplanes in various categories must be designed, and he is to be thanked by all who are interested in these problems for the time and care he has devoted to the presentation of these suggestions.

Control by Reference to Angle of Attack

RALPH H. UPSON[4]:—To hear a paper that goes back to real fundamentals is certainly a great pleasure. The only excuse for discussing such a paper is that there are fundamentals and fundamentals. Sometimes, in tracking a thing down, when we come to a fork in the road, as in this case, there seem to be three or four prongs to the fork.

I decidedly disagree with both the author and Professor Newell as to the proper criterion for most commercial types of airplane with respect to maneuvers. It seems to me that neither the stick force nor the elevator angle is the proper criterion to use. Anyone who drives a car can wreck it at any time by over-application of the steering control, in which neither the force applied nor the amount to which the steering-wheel is moved is the practical criterion. What is the criterion? It is something like this: You obtain by experience the "feel" of the car when it is first starting to skid, you coordinate with that the sensation of a lateral centrifugal force which usually is sufficient to produce such a skid, and you subsequently refrain from any maneuver severe enough to produce that sensation.

Unfortunately, when in the air we cannot wait until the machine starts to disintegrate before refraining from pulling back farther on the stick, but we do have a very distinct advantage in the air, which is the fact that the air is of much more uniform texture with respect to the effect of a given angle of attack than is the road to lateral forces on the automobile. If the pilot has no other means of judging, he can judge by his own "feel" of the load factor.

Anyone who wants to practice the feel of a certain load factor can go to an amusement park or playground and swing up to an angle that is straight out from the pivot of the swing, in which case he will get exactly an acceleration of $2g$ at the bottom, or a total load factor of 3. He will find that it is very definitely to be felt, and, with a little practice, he can obtain a fairly good approximation of when he gets such a load factor. Although such judgment based on feel is admittedly crude, I believe that most pilots would agree that the direct feel of the load factor itself is more dependable than trying to go by the force or position of the stick, especially on widely different types of aircraft.

As is shown in my paper on the Air-I-Zon[5], a really accurate control of load factor is more readily obtained through reference to angle of attack than it is by the load factor itself. In this way, practically any limitation of load factor short of that necessary to avoid hitting an obstruction may be obtained very readily, so I see no excuse for any machine being arbitrarily controlled beyond the limits of its own structural safety.

The gist of the whole matter seems to me to be simply this: that we are coming more and more to regard the atmosphereic gust as the fundamental factor in the allocation of load factors.

Stresses Imposed by Vertical Gusts

MAC SHORT[6]:—We unfortunately encounter considerable bumpy air in the Central States, and often wonder about its significance. I should like to learn if the operators of transports believe that the values used by Mr. Rhode and later by Mr. Warner approach the maxi-

[4] M.S.A.E.—Aeronautic engineer, Ann Arbor, Mich.

[5] See S.A.E. JOURNAL, April, 1932, p. 159.

[6] M.S.A.E.—Chief engineer, Stearman Aircraft Co., Wichita, Kan.

mum that would be encountered. The question is prompted by the experience of a transport operator who encountered a bump that resulted in an indicated rate of climb of 1000 ft. per min. with the plane nosed down slightly and with the engines well open. Probably it was not a line squall but a bad mountain storm. Further, when one strikes such a bump, what is the correct procedure to follow to make the stress the minimum on the structure?

RICHARD V. RHODE[7]:—I do not feel qualified to tell Mr. Short what to do to relieve the stresses in the airplane when he encounters one of these bumps. As a rule, the gusts that are most serious from a structural viewpoint have such a sharp gradient of velocity in their horizontal cross-section that I am sure the maximum stress would be experienced as soon as the pilot became aware that he had struck a bump. Probably the pilot cannot do much about it.

The indicated rate of climb of 1000 ft. per min. checks rather well with some of the values we have measured. It is not regarded as an unusual figure although, if it continues for a considerable period, it may be quite impressive to one riding in the airplane. With respect to the stresses imposed by the gusts, the absolute value of the vertical velocity, while important, is not the whole story. The velocity gradient or cross-section of a gust is perhaps of equal importance, because, if it has certain values, it may be a contributing cause of overstress. The wings of an airplane encountering a gust receive the load first and are immediately deflected upward with respect to the fuselage. If the velocity gradient of the gust is steep, the wings will accumulate energy in the sudden upward deflection which must be absorbed as internal work in the structure, and the result is what we might term a dynamic overstress. That is, if we measure an acceleration of, say, 2.5g on an airplane when it encounters a gust, the actual stress in the wings, particularly of the larger transport planes whose wings are relatively flexible, actually will exceed the stress corresponding to an acceleration of 2.5g or a load factor of 2.5.

Küssner in Germany has made an exhaustive analysis of stresses encountered in gusts and has calculated the dynamic overstress for a number of existing German airplanes. He finds that, with gust gradients which seem to exist, the dynamic overstress, or the ratio of the actual stress to the stress corresponding to the acceleration considered as a static load, may be as high as 1.7; the exact figure varies with the cross-section of the gust and the flexibility of the wing structure.

JOHN D. AKERMAN[8]:—Two aspects should be considered before accepting a method for determining load factors for airplanes. The first occurs when the load factors should be determined for design purposes before the plane is built and its characteristics are determined by a flight test; the second occurs when a plane has to be analyzed after all flight characteristics are known.

It is very desirable to be able to determine the necessary load factors for an airplane before all flight characteristics are known. The Department of Commerce so far has very wisely used a method which would not require any data that are not known to the airplane designer from the start. If we could in the future follow the same principle, the designer and the analyst would be greatly helped and time and money would be saved for the manufacturer.

RICHARD C. GAZLEY[9]:—I wish to echo Mr. Wright's appreciation of the time and effort that Mr. Warner has put on this subject, as I am sure all the rest of you do. It is a very complex problem and any efforts that are directed toward solving it are greatly appreciated.

I am sure, however, that Mr. Warner did not intend to create the impression made by the statement in the paper to the effect that the British and American authorities appear to be satisfied with the existing design-analysis conditions. He doubtless knows that the Department of Commerce, together with many of the aeronautic engineers of the United States, has been working for at least a year on the development of more rational design methods. I might go farther back than that and say that the apparent satisfaction which existed prior to a year ago was not really satisfaction with the existing conditions but was merely inability to do much about it because of lack of basic aerodynamic data. Even now much remains to be learned concerning the actual loads imposed on the airplane and regarding this gust problem before a regulatory basis for stress analysis can be set up which will have any assurance of being better than the one we have at present. This additional information which Mr. Warner has put forth will be welcomed and studied in the program which we are now carrying on in Washington.

Easier To Gage Stick Force than Stick Movement

MR. WARNER:—Professor Newell questioned the validity of the constant stick-force and thought that elevator angle was better as a basic measure of specification. I believe that both should be taken into account. In the case of balanced controls, we probably shall come to some limiting angle as a supplementary specification, even if stick force should generally be taken as basic; but it seems to me that we are in some danger of being misled by the result of experiments undertaken for the specific purpose of finding out how large a load could be imposed upon an airplane. In those cases the machine was dived usually at a speed substantially below the limiting speed of vertical dive—somewhere in the neighborhood of 200 m.p.h. in the case of pursuit planes—and has been pulled out as abruptly as possible. This might mean that the stick force might be a little below the 100 lb. but that the elevator would be pulled up sharply to its limit of travel.

That seems much less likely in the relatively steep dive and quite unlikely in normal maneuvering. I think the pilot, in coming out of a steep dive, is likely not to move the elevator more than 6 or 8 deg. even in applying the stick force that I have suggested, which means that he would be moving the head of the stick only perhaps 2 or 3 in. It is much easier for him to gage the force he is applying than to gage so small a movement.

Feeling of Acceleration a Poor Gage

I find it easy to agree with Mr. Upson up to a certain point, but all pilots are not gifted with his talent for analyzing things while they happen. He said that the automobile driver gages the making of a curve by the lateral acceleration. That is true for 1 automobile driver out of 25; most drivers gage the turn by the feeling of speed and the fact that they can see the radius of turn, which one cannot do in the air. They note the sharpness of the turn and regulate the speed accordingly; and, if a skid starts, they become alarmed and do not know what to do except to apply the brakes, which makes matters worse. I fear we shall have many such persons in aviation.

I agree that load factor does play a certain direct part in specification, especially for very high factors. Pursuit pilots undoubtedly depend on it. In general, however, that is a supplementary item, not the major one or the one most generally applicable, and I believe

[7] Associate aeronautic engineer, National Advisory Committee for Aeronautics, Langley Field, Va.

[8] Professor and head of department of aeronautical engineering, University of Minnesota, Minneapolis.

[9] M.S.A.E.—Chief, engineering section, aeronautics branch, Department of Commerce, City of Washington.

it is not very applicable in large machines where the load factors are likely not to exceed 4 or thereabouts in any case. Suggestions have been made that load-factor indicators be applied on the instrument boards. The Germans have done it to a certain extent. They have a little accelerometer that flashes a light at a certain point, but this is a secondary method, and I doubt if the pilot will watch it at the critical moment.

German Gliders Soar in Gusts

Mr. Short cited the climb of 1000 ft. per min. in a gust. That is only 17 ft. per sec. After the plane gets into the established condition of the vertical current, the air is lifting the airplane directly at its own speed, or is modifying the rate of climb by its own speed, so Mr. Short's figure lies well within the limits that have been assumed for comparatively normal conditions of about 30 ft. per sec. The Germans probably have more direct measures on that point than anybody else, because pilots of German gliders having a 60-ft. span and no powerplants have, with amazing intrepidity, deliberately played about in and out of thunderstorms for hours, and they get incredible rates of climb in some cases.

Mr. Rhode's point on dynamic overstress in wings is of extreme interest. I must admit not having had it in mind until Mr. Rhode called it to my attention yesterday. Although it should have been obvious enough, probably nobody else in the Country has had it in mind. The German figures presumably are authentic. I should suspect, however, that in most cases this again would be a rather small factor; first, because the elasticity of the airplane wing is small enough so that the time required for it to deflect would be short compared with the time required for the passage into the gust to be complete; second, and I suspect this amounts to considerable in some cases, especially on riveted metal planes, because of damping in the structure itself. Any deflection will require, not only elongation of compression members, but a considerable relative displacement of certain members, and I imagine that the damping would be considerable.

Finally, Professor Akerman's suggestion is that we should not require an N.A.C.A. flight test on every type of airplane. I absolutely agree with him and I thought I had carefully indicated it by saying that I did not want to take maximum speed into account but only those things that are obvious from the assembly drawing of the airplane.

As regards Mr. Gazley's suggestion that I was unintentionally critcal of the Department of Commerce, I can only express regret if it appeared that way. The sentence he cites was merely a rhetorical flourish, meaning that neither the United States nor England had admitted that the other was right.

Why Gust Rate Is Used as a Factor

LIEUT.-COM. R. D. MACCART, U.S.N.[10]:—Whether we use 30 or 25 ft. per sec. or any other number of feet as the rate of a horizontal gust makes very little difference. We find that 30 ft. per sec. is equivalent to about $4g$, which we do not regard as an excessive acceleration in low-incidence inverted flight; that is the real reason that we have adhered to the rate of 30 ft. per sec. On top of that we put a factor of safety, which ordinarily is 1.35, based on yield point. We found that it was very easy to convince designers, on the basis of gust, of the necessity for more strength in inverted flight, and we used that means rather than a more arbitrary means to increase the inverted-load factors.

W. A. KLIKOFF[11]:—I do not know whether there are any data to support the horizontal-gust rate of 30 ft. per sec. As far as I am familiar with the atmospheric conditions, I do not think that sudden horizontal gusts of this velocity can be encountered.

Mr. Warner uses a factor of 100-lb. force on the stick as a design criterion. Therefore all the factors of the weight of the plane depend upon the leverage from the control stick to the tail surfaces.

Another point is the question of balance and the size of the tail area. We know that we may design a plane having a certain tail area, a certain amount of balance and a certain leverage, but during the actual construction of the plane or after trial flights these may all be modified to such an extent that the original layout is virtually obsolete. Perhaps that is the reason why Mr. Warner started from the fundamentals. But in the matter of stick force he has approached the general design and has somewhat departed from fundamentals. I cannot offer a solution, but I suggest that the principle of using the stick force as a design criterion could be modified.

LIEUTENANT-COMMANDER MACCART:—I feel that a rate of 30 ft. per sec. is not too severe for high-performance diving planes. The British authorities state that they have increased the inverted-load factors and are giving serious consideration to the question of just how much the increase should be. Although we use an equation based on Mr. Rhode's paper[12] showing that a velocity of 30 ft. per sec. is to be anticipated, we have more or less submerged it after considerable experience and now use a formula that includes 30 ft. per sec., or whatever velocity we wish to specify, from which we obtain the inverted-flight acceleration. We have made some inverted-flight tests and measured inverted acceleration in reasonable maneuvers approaching $4g$, and for that reason we generally specify $4g$ as the minimum inverted and positive accelerations. It seems consistent to use the equivalent of 30 ft. per sec.

Ultimate Load Factor Might Be Smaller

MR. WARNER:—In the case of inverted loading, which I did not touch upon in the paper, I am disposed again to take two distinct points of view, to approach quite independently the inverted loads in maneuvers and inverted loads due to gusts. We can have downdrafts approximating in velocity the updrafts that have already been mentioned, and we know we can get very large negative load factors as the result of a sudden nosing over from a gentle descent at full power. From what little work I have done on the subject, I should say that my own conclusions would lead to a considerably smaller ultimate load factor than would be given by the specification that Lieutenant-Commander MacCart has laid out, including a factor of 1.35 based on the yield point.

The horizontal gust of 30 ft. per sec. during a dive has been spoken of several times, but Mr. Klikoff questions the existence of any such gusts. It seems to me that we are rather leading ourselves astray there, or that we are using a misnomer. What we are really concerned with, as I understand the problem, is not a true horizontal gust but a change in wind velocity. A pilot diving in a strong wind velocity does not remain in a stratum to investigate the gusts; he goes to new levels and new conditions. A machine diving at 250 ft. per min. changes its altitude 1500 ft. in approximately 4 sec., and, under some conditions, the pilot can get a wind change of 30 m.p.h. in a time that is so short that the machine cannot accommodate itself to the changing conditions by natural rotation in space.

I am disposed to sympathize with what Mr. Klikoff says about the excessive stress thrown on the control leverage. The designer can control that if he insists

[10] In charge of airplane structures section, Bureau of Aeronautics, Navy Department, City of Washington.

[11] Aeronautical engineer, Metalclad Airship Corp., Detroit.

[12] See S.A.E. JOURNAL, September, 1931, p. 179.

upon doing so, and if it is a real factor in the loading of an airplane we should be prepared to control it.

I have the same feeling about one other element the omission of which from the paper may have caused some surprise, especially at the Massachusetts Institute of Technology; that is, the stability characteristic of the plane. This has a very pronounced effect. A stable airplane cannot be maneuvered with as much rapidity nor subjected to such high load factors as a result of maneuvering as can one that is unstable or neutrally stable. Most of our airplanes are stable. The figures included in my paper are all based on neutral stability, not only for the very practical reason that this was the most simple thing to do, but because I thought that this was the widest assumption we are justified in making. There always is a possibility of a shift in loads which will move the center of gravity back enough to change stable characteristics a very appreciable amount. It does not seem to me justifiable to rely on the existence of positive longitudinal stability to reduce loads when it may so easily be substantially reduced, if not destroyed, by changes made in the loading of the plane.

Bodies Considered from the Car Buyer's Viewpoint

(*Concluded from p. 294*)

what offended by the tinny appearance of some of the fittings and are demanding ash trays, robe rails and foot-rests of better quality than they have been getting. In the medium-price class, the dome lamp which lights when the door is opened is a great selling asset.

The divided back seat and, in the higher-priced cars, the divided front seat often add just enough to the interest of the buyer to make him want to take delivery at once.

More Comfort Being Demanded

With reference to comfort, the day is past when we can sell in any price class a car that has a very thin cushion and coil springs that can be felt through the padding of seats that do not have the correct contours. We have not as yet offered the public as much in comfort as it is going to demand in the next few years. Comfort is a car factor that automobile designers can and must provide, and the manufacturers whose engineers develop greater comfort in the automobile will give their sales departments much assistance. The demand in the last few years for low bodies and double-drop frames has resulted in a sacrifice of head-room for appearance. The public is rapidly coming to the frame of mind to demand more head-room and leg-room, a back seat that will accommodate three persons in a so-called five-passenger car, and doors that are wide enough to enable a person to get in and out without turning sideways.

Body types most demanded by the public are the four-door sedan in all price classes, the two-passenger coupé in all prices classes and the two-door sedan in the lower-price class. Convertible bodies are rapidly increasing in popularity, and a strange fact is that the market for these convertibles seems to be in the New England States and the Central West, and not in the South or on the Pacific Coast. The reason is that the hot sunshine in the South and in California not only bleaches the top material, but beats through it and makes the convertible car much hotter than an ordinary sedan or coupé.

Car buyers would be interested in many different body types, but if the dealer is to make money the number of body types must be limited, as, with his present limited capital, the dealer cannot carry too large a line or, rather, any slow-moving merchandise. We should give him body types that make possible a fast turnover for quick profits; many manufacturers would have a happier and more prosperous dealer body today and do a better business with fewer models.

Distribution of Oil in the Engine

EXPERIENCE with racing engines has demonstrated the advantages of supplying a large quantity of oil at low pressure to the bearings rather than a smaller quantity at high pressure. The quantity of oil circulated should be the most that a well-designed aluminum piston fitted with one single-duty oil ring and one high-unit-pressure oil-control ring and operating in a good bore can control to an economy of 1000 to 1500 miles per gal. of oil at 50 m.p.h.

Distribution plays an important part in efficient lubrication. An oil manifold supplying oil to the bottom of the main bearings is recommended, for the sake of supplying lubrication under pump pressure, instead of only centrifugal pressure, for lubricating the cylinder-walls in starting and for the piston-pins.

The life of the pistons and cylinder bores depends upon an immediate supply of oil in the cylinder bore when the engine starts cold. This is best provided by direct pressure in the following way:

The oil is introduced into the lower half of the main bearing through an oil manifold, an oil groove being provided that extends at least 60 deg. on each side of the oil hole. The oil is led to the connecting-rod bearing through a hole in the crankshaft and sprayed into the cylinder through a hole in the connecting-rod bearing registering with the hole in the crankshaft. The latter hole should be on the trailing side of the rod and located so that its projection will strike about ½ in. from the edge of the cylinder bore at the instant when the two holes cease to register. The spray hole should be about ⅛ in. in diameter for approximately 3/16 in. at the bearing side and about 0.05 in. for the rest of the distance.

With this arrangement, the oil will be projected to the edge of the bore at 600 to 800 r.p.m. As the speed increases, the trajectory of the spray will change to bring the oil first higher in the bore and then clear of the bore on the opposite side. A correctly located oil hole will not cause high oil consumption and will provide the difference between success and failure of the piston installation.

In some engines, the oil is supplied to the upper half of the bearing in such a way that the pressure from the pump is shut off at the time when the squirt is needed from the hole in the connecting-rod. Centrifugal force cannot be depended upon to supply sufficient force for this at the low speeds at which the oil in the bore is most desirable, particularly when the viscosity of the oil is low.

With a bushing at the upper end of the connecting-rod, which is now almost universal, pressure lubrication through the connecting-rod is found most satisfactory both in service and manufacture. To be assured of sufficient oil under pressure, the rifle-drilled hole in the rod must register with the hole in the crank while the oil therein is subjected to pressure from the pump. This also can best be accomplished when the oil is introduced into the main bearing at the bottom.—From Annual Meeting paper by D. E. Anderson, M.S.A.E., of the Bohn Aluminum & Brass Corp., Detroit.

Use of Tractor Equipment on the Mississippi River Levees

By C. E. Frudden[1]

Milwaukee Section Paper

MAGNITUDE of the Mississippi River flood-control work and the huge quantities of earth that must be moved in the building of levees in the South require tractors of the largest commercial sizes and varied dirt-moving machines and wagons to perform the work expeditiously and economically. The author describes briefly the types of tractor and tractor-operated equipment used by the contractors as he observed them at work on two visits to the South.

How several hundred tractors are regularly used for hauling crawler-track dump wagons, pulling blade graders and elevating graders, pushing bulldozers and supplementing tower machines and drag-lines is described, and the author pictures vividly the severity of the work, which requires continuous operation for 20 to 22 hr. per day, seven days per week, and the crude provisions for and methods of maintenance, as a consequence of which the machine casualties are heavy.

As the engineers who design and build this automotive equipment seldom have an opportunity to see it under such operating conditions, the paper may supplement their present information and aid toward improvements in mechanical designs.

ALTHOUGH a large number of automotive engineers are engaged in designing and building dirt-moving machinery or automotive engines used in this equipment, the last that we engineers see of either as a rule is when it leaves the paint shop. In time we may hear about some weaknesses through complaints to our service departments, but we know very little of the whole problem in which a large part is played by the machinery we design and build.

On two occasions I have been down South to observe the product of the company with which I am connected at work on the flood-control levees of the Mississippi River and, while visiting the camps of the various contractors, was impressed with the frequency with which such well-known names of Wisconsin manufacturers as Allis-Chalmers, Fairbanks-Morse, Koehring, Le Roi, Monighan, Harnischfeger, Northwest, Waukesha and Wisconsin are noted on the equipment. Some brief description of the equipment and remarks on how it is used may add to the information we already have and perhaps find its way eventually into improved mechanical designs.

Our company builds tractors, elevating graders, crawler wagons and blade graders that constitute a line of dirt-moving machinery which sometimes is used as a complete outfit and often in conjunction with other well-known equipment.

Tractors Used in Levee Building

The first impression one receives upon getting close to the levee construction work on the Mississippi below Memphis, Tenn., is the immensity of the job. The work has been going on for 50 years and still is only beginning. It consists in reinforcing and building up old levees, building new ones to provide a wider river channel with higher banks and building up the banks of the numerous laterals or bayous. In most cases this involves the simple work of digging up nearby dirt and building a high bank to confine the river water at flood stages, but its magnitude makes it an engineering problem.

Manufacturers of dirt-moving machinery find plenty of competition among themselves in developing apparatus to reduce by 1 or 2 cents per cu. yd. the cost of moving the millions of cubic yards involved. The automotive type of gasoline engine is seen at every camp, installed in numerous kinds of equipment, including lighting plants, portable arc-welders, power shovels, drag-lines, tractors and special machinery. These engines work long and hard and are treated none too well. This paper will be confined to the part that tractors are performing in this program.

Several hundred tractors are regularly at work on this levee project. Only the larger commercial sizes seem to have found a place for themselves. They are of the track-laying type weighing 10 to 12 tons and powered with engines delivering 80 to 100 hp. Levees built entirely by tractor power have the reputation of being good, and it has been stated that no tractor-built levee has ever been washed out. This is because the tractor must continually pull itself and heavily loaded wagons over the dumped earth and in doing so packs the dirt most effectively.

Placing Dirt on Levee Top

No matter what sort of equipment is used for actually placing the dirt on the levee, tractors do plenty of work in clearing the land of trees and stumps, as shown in Fig. 1, hauling supplies over the impassable roads and the like. Finishing of the levee slopes is almost universally done with tractors pulling blade graders or with bulldozer equipment. However, the big job is moving the dirt to the top of the levee. In this work the tractors, each pulling one or more wagons of 7 to 8-cu. yd. capacity, as shown in Fig. 2, either supplement the large tower machines and drag-lines or do the work in competition with them. When used on very large projects to supplement the large tower machines, or in conjunction with relatively small drag-line outfits, the tractors may open up the far side of the borrow pit and build the base of the levee. With this preliminary work done, the tower machines can work to their best advantage.

Supplementing the large portable drag-line outfits having booms that are not long enough to complete the dirt handling in one operation, tractors and wagons are used to deliver the dirt from borrow pits to a point within reach of the drag-line bucket. (See Fig. 3). The drag-line picks up the dirt dumped from the wagons and spills it on top of the levee. This system relieves the tractors of the hardest work of pulling themselves and the heavily loaded wagons up the soft dump to the top of the

[1] M.S.A.E.—Chief engineer, tractors, Allis-Chalmers Mfg. Co., Milwaukee.

levee, the operation being greatly speeded up and the wear and tear on tractors greatly reduced.

In other cases tractors with wagons haul dirt from the far side of the borrow pit and build the base of the levee, the top being completed by the drag-line, which moves the dirt that is within its reach.

A Completely Tractorized Outfit

Building levees exclusively with tractor power is, of course, the method recommended by the tractor man. A fleet of four or five tractors, each pulling one or more 7 to 8-cu. yd. crawler-type wagons, is the common dirt-hauling outfit. Sometimes a drag-line or power shovel is used to load the wagons, but the completely tractorized outfit has a tractor pulling an elevating grader for loading the wagons and another tractor equipped with a bulldozer for leveling the runways and the dump. Such an outfit is very much like that commonly used in road-building operations, but the levee work may be harder because the loaded wagons must be pulled up the soft dump, whereas in road-construction work the load is generally pulled down hill to the dump.

The bulldozer, which is a valuable tool for the levee contractor, is a big plow carried on the front of the tractor, as shown in Fig. 4, and is raised and lowered by power under control of the tractor operator. It maintains the runways to the top of the levee in good condition for the tractors to pull heavy loads at higher speeds and avoid delays caused by becoming mired. With a bulldozer on the job, the tractor-drawn wagons need not be run dangerously close to the edge of the dump. The loads can be dumped where convenient and the loose dirt pushed by the bulldozer to the desired spot.

The elevating grader is used for loading the wagons and is pulled by a tractor, as in Fig. 5. It is fitted with a disc plow 26 to 32 in. in diameter, which cuts a slice or furrow 18 to 20 in. square in cross-section and deposits the dirt on the moving elevator belt, which carries it to the wagons as they are pulled alongside by other tractors at a speed of 2 to 3 m.p.h. The elevating belt or conveyor is driven either by the tractor's own power by a power-take-off shaft or by an independent gasoline engine mounted on the grader. The latter scheme relieves the tractor engine of considerable work and makes the full tractor power available for pulling the grader and plow, but, on account of the additional equipment involved, this very good idea has not yet been generally adopted by the levee contractors. Dirt can be loaded

FIG. 1—TRACTOR CLEARING LAND OF TREES AND STUMPS

This Work Includes Clearing the Borrow Pit and the Levee Base Preparatory to the Actual Moving of Dirt for Building the Levee

FIG. 2—HAULING DIRT TO TOP OF LEVEE NEAR GREENVILLE, MISS.

The Tractor Is Hauling 14 Cu. Yd. in Crawler-Track Wagons, Which Are Fast Superseding the Wheeled Type. Lamps on the Tractor Supplement the General Lighting System for Night Operation

FIG. 3—DRAG-LINE LOADING TRACTOR-DRAWN CRAWLER-TYPE DUMP-WAGON

Tractors Haul to the Drag-Line Dirt That Is Beyond Its Reach, and the Drag-Line Relieves the Tractors of the Hardest Work of Pulling the Loaded Wagons to the Levee Top

FIG. 4—BULLDOZER MOUNTED ON AND OPERATED BY A TRACTOR

This Is Used for Leveling Loose Dirt and Maintaining Runways for Tractor and Wagon Outfits

into wagons with elevating graders perhaps at lower cost than by any other commonly used method, provided that ground conditions are favorable. In loose, sandy soil or where pits are wet, the drag-lines have certain advantages.

The wagons now generally used are carried on crawler tracks similar to those on the tractors and are provided

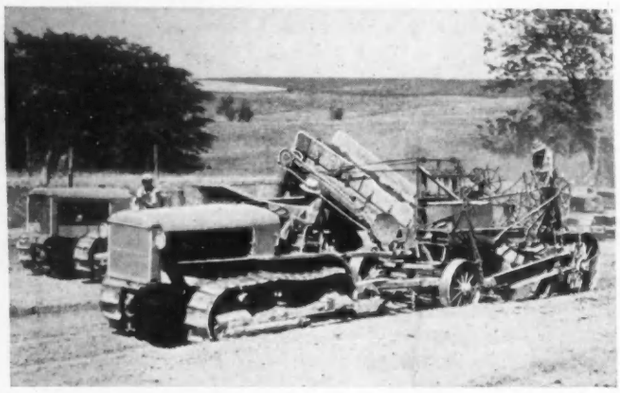


FIG. 5—A COMPLETELY TRACTORIZED OUTFIT

A Tractor-Drawn Elevating Grader Loading One of Five Tractor-Drawn Crawler-Type Wagons Hauled Alongside

with dump bottoms. The earlier wagons, fitted with large wheels, pull harder and mire down more easily than the modern type.

Standardization of power in the case of completely tractorized outfits is an attractive feature. When all of the tractors are of the same size and type, one extra tractor assures freedom from breakdown delays, and the stock of parts needed for repairs, which at best are made with difficulty in these camps, is reduced to the fewest possible items. The versatility of the tractorized outfit that is suited for all jobs, large as well as small ones, whether building new levees or enlarging old ones, makes it a good investment, and the flexibility of the relatively small units makes the moving from one job to another an inexpensive item.

Operating Conditions Make Machine Casualties Heavy

Levee building is carried on throughout the year, working day and night, Sundays and holidays, except that during periods of high water, when seepage is excessive, all of the outfits find it difficult to work efficiently. One of the levee man's problems is to keep his pits in such shape and his work so organized as to provide suitable drainage of water to low places so that, even after heavy rains, he will quickly have dry ground on which to continue his work with the minimum delay.

Operating 24 hr. per day with two shifts is the rule. The tractors average 20 to 22 hr. of work per day, the lost time being accounted for by time out for meals, filling fuel tanks, greasing and the like. The tractors actually work 500 to 600 hr. per month and up to 5000 hr. per year. In comparison, a farm tractor usually does not operate more than 500 to 600 hr. per year, and road-building tractors average 1500 to 2000 hr. per year. Motorcoach or motor-truck operation for 5000 hr. at say 30 m.p.h. means 150,000 miles. Obviously, therefore, this levee work calls for good equipment.

Negro boys operate the tractors and become very efficient. White men maintain the tractors and other equipment but their facilities for working with engine bearings, cut gears, ball bearings, magnetos, carbureters and other parts are very crude. I never have seen a repair shop enclosed and under roof in a contractor's camp. Sometimes a canopy protects the men from the hot sunshine, but everything becomes covered with dust in dry weather and with mud after every rain. Sledge hammers and crowbars are the tools most used.

As the levees are built on schedule, the equipment casualties run high. A condition making bad matters worse for the machinery in need of repairs is that all contracts call for a certain minimum amount of levee built every month, to ensure completion of the contract at the prescribed time. Every job must be completed within the season, and the Government engineers do not hesitate to call in outside help to bring a lagging contract up to schedule. Profits disappear quickly under such "force-account" work, therefore the temptation is great to keep the equipment continuously at work until it quits entirely rather than to lay it up for minor repairs or regular inspection.

Driver Accident Repeaters

REPORTS on motor-vehicle drivers who were involved in two or more accidents at Pittsburgh during the four years from 1928 to 1931, inclusive, have been analyzed by the Bureau of Traffic Planning of the City of Pittsburgh and a report on the subject has been prepared and issued by Lewis W. McIntyre, traffic engineer. This shows that 371 drivers were involved in two or more accidents during the four years, of whom 25 were involved in three and 2 drivers in five accidents. The 371 drivers constituted slightly less than 2 per cent of all drivers concerned in accidents, and they were involved in 773 accidents, or 6.4 per cent of all those reported.

In the four years, 137 drivers had two accidents within two years, 199 within three years and 318 within four years. Nine motorcoach drivers and only one woman driver were among the 27 who had three or more accidents. Of the 27, 15 seem to be deficient in safe-driving knowledge and apparently are in need of further examination prior to licensing, but the remaining 12 were involved through no fault of their own.

The Bureau of Traffic Planning states that no indication of a trend to increased accident proneness on the part of drivers is shown, that the accidents reported by motorcoach drivers were mostly due to passengers being shaken up in the vehicle or in getting on or off and that none of the accidents involving drivers with three or more accidents resulted fatally.

The Traffic Observers, an organization sponsored by the Mayor's Better Traffic Committee and composed of 500 citizens of Pittsburgh, reported 5667 serious traffic-law violations between May 15, 1930, and Dec. 31, 1931. Among the drivers cited, 56 repeaters were reported 119 times, or slightly more than 1 per cent of all drivers were responsible for slightly less than 2 per cent of the reports. Five of the repeaters have been involved in accidents.

Types of violation reported included weaving driving, comprising 16.0 per cent of the reports; too fast for safety, 15.1 per cent; did not obey Stop and Go signal, 14.8 per cent; passed to left of street car, 13.4 per cent; reckless driving, 9.0 per cent; and did not yield right-of-way, 8.0 per cent.

Conclusions reached from the analysis are that, in at least 50 per cent of the accident cases, the repeater seemed responsible; that drivers reported repeatedly for traffic-law violation arouse a strong suspicion as to their driving fitness; and that reexamining repeaters and removing them from the road if such examination shows them to be incapable of proper driving seems logical.

Prevention of Motorboat Fires and Explosions

Metropolitan Section Paper

By James Craig[1]

THE subject of fire prevention aboard the modern motorboat has been carefully studied by competent people, and their deliberate conclusions are presented through the National Fire Protection Association in Appendix D of Regulations Governing Marine Fire Hazards for Internal-Combustion Engines. These regulations have been approved and are recommended by the National Association of Engine & Boat Manufacturers.

My purpose is to present herein some conclusions from experiences and observations of motorboating that may emphasize the importance of these regulations.

Our endeavors to bring about greater safety aboard motorboats may be helped if we concede that, while a fire thereon may present a serious phase, a human being may retain possession of his ability to think and act wisely in the emergency, whereas, if an explosion transpires, overwhelming disaster may result.

Fires and explosions are in many instances the consequence of the ignorance or neglect of human beings in the handling and use of combustibles. This human deficiency is too often involved in the operation of a motorboat.

Motorboat design often provides unavoidable confines that are difficult to clean and to ventilate and that readily become filled with leakages of dangerous combustibles. Motorboats are intended for cruising and adventuring on navigable waters far from communications or any emergency base; and in such isolation a minor mishap may sometimes lead to a serious end. Therefore we should pledge our earnest endeavors to work for every improvement that will minimize the danger.

The aforementioned regulations governing marine fire hazards are very helpful, and the Department of Commerce Circular No. 236, June 1, 1927, prescribes a Regulation of Motorboats and specifies the equipment that is mandatory on them.

Precautions when Filling Fuel Tank

Filling the gasoline tank aboard a motorboat creates a dangerous period. Profound concentration should be exercised while performing this operation and I advise motorboat management to insist that no one be allowed on board except those required for it. I prefer a filling station located on shore, outside the zone of disturbance and turbulence, and provided with convenient servicing and emergency facilities.

A clear, dry day attended with a breeze is most helpful for diffusing the combustible gases displaced from the fuel tank. A hot, humid day with no atmospheric disturbance allows these released gases to settle and accumulate in confined spaces, displacing lighter gases and thereby creating an explosion hazard.

All working operations aboard boat should be stopped during the filling operations, and all ports, hatches and other apertures should be closed before filling the gasoline tanks, to preclude the admittance of any uncontrolled combustible gases. After filling operations are wholly completed, the ports and hatches should be opened and an effective air circulation induced throughout the vessel, thereby dispersing any contained combustible gases. Next, a deliberate inspection of the vessel should be made in an effort to find a condition that never occurred before and correct it before it phenomenally creates a serious hazard.

All volatile combustible liquids should enter the tank by passing through the liquid at the bottom. This system minimizes turbulence and the release of an unnecessary quantity of gases.

Where it is permissible I prefer a gravity feed of fuel from the tank to the engine, because of its simplicity and the absence of mechanism. I also recommend that the tank outlet extend within the tank in the form of an inverted trap, in which the discharging liquid must ascend for a distance. Such a device allows gravity to act in the vertical lift therein, and if the trap orifice is large enough to effect a desirable minimum fuel velocity, no foreign matter can pass through the outlet into the engine.

Fuel tanks should be provided with suitable vent pipes of adequate size, so that in the event of a fire, if the fuel tank is exposed to undue heat and becomes a gas generator, the gases will be gradually released to the outer atmosphere and the tank structure relieved of all internal gas pressure that otherwise would rupture it and suddenly release the explosive contents.

Galley Stoves a Source of Danger

Galley stoves are passively conceded to be the most backward detail on board a motorboat, and sometimes they are forcefully condemned. They are made in many varieties for using every kind of fuel, notwithstanding there is but one best fuel for the purpose. Many of these galley stoves are arranged with burners that require preheating by a volatile liquid combustible, which entails additional gadgets. Most of these mechanical devices are liable to mismanagement and their failure to function properly is troublesome and hazardous.

The galley range using bottled gas for fuel is conceded to be the superior cooking device. The gas ignites instantly, combustion is complete and clean, regulation of the flame is very simple, and it does not soot or smear the cooking vessels. But all of these highly commendable properties are sorely hampered by the fear of a leakage of this highly dangerous bottled gas. The leak must be stopped to cure the fear. I believe that the fittings and connections in this system can be improved so that the galley gas range will equal in service and safety the standard home gas range. I suggest that they they all be made of forged or rolled material; no castings, with their inherent porosity, should be countenanced. These fittings should be made liberally robust and capable of enduring abuse. All tube fittings should be specially made and should be superior to the S.A.E. Standard Practice code type; in fact, I think that this standard code type is not good enough to be used anywhere on board a motorboat.

This contemplated galley gas range should be complete with a new special outfit of cooking utensils. They should be arranged with convenient devices whereby they can be affixed to and released from the range by a partial turn of the utensil or an equivalent motion. Turbulent waters often and unexpectedly project a pot or stewpan with its hot contents to the floor.

[1] Formerly advisory engineer to technical committee, National Association of Engine & Boat Manufacturers, New York City.

A critical unsafe period aboard a motorboat is after the boat has been sealed for a day or two and then hastily opened and started. While the boat is in motion effective air circulation is maintained in and about the compartments and confines and, if a small leakage of gasoline prevails, the gas is quickly and effectively diffused. But when this circulation ceases and the stored heat of the engine is radiated, a condition of extreme hazard is created. The best cure is to stop the leaks and maintain air circulation.

Fundamentals of Automotive Lubrication

(*Concluded from p.* 282)

information appears to be available on this point.

In the case of a journal bearing, the load is distributed over a fairly large area. With ball and roller bearings, it is true that the load is distributed over a number of balls or rolls, but the area of contact of each ball or roll with the race is small, with the result that the load per unit area of contact is much larger than with journal bearings. If the relative motion were all sliding, this increased load per unit area would make the matter of wear more important with antifriction bearings than with journal bearings. However, since the relative motion is largely rolling, wear with ball and roller bearings is not a very important factor under ordinary operating conditions if the bearing is suitably protected from abrasive materials. Under the excessively high loads which are coming into use, however, wear may become very important. With a properly designed ball or roller bearing, suitably installed and operated under normal conditions, the main consideration appears to be the length of time the steel will resist fatigue failures. Use of high-tensile-strength steel and special methods of heat-treatment have resulted in long bearing life.

Thus it appears that, under normal conditions of operation, the problems of a practical nature connected with antifriction bearings are minor. Under high-speed or high-load operating conditions, the problems of friction and wear may become of great importance, but few experimental data are available.

Characteristics of Gears

The relative amount of sliding and rolling with gears varies all the way from almost complete sliding with a worm gear to almost complete rolling with a spur gear. With a 45 deg-spiral gear arranged to transmit the motion through 90 deg., the relative amounts of sliding and rolling are about equal. Friction itself is of little interest in connection with gears except insofar as it gives an indication of the temperatures that may result at the gear teeth. Because of the very high loads carried by some types of gear, such as worm and hypoid gears, the matter of wear and seizure is of the utmost importance. Special extreme-pressure lubricants are necessary in many cases to prevent rapid scoring of the metallic surfaces which, if not prevented, would soon lead to seizure. While little information is available at present on wear with gears or on the use of extreme-pressure lubricants to minimize this wear, the widespread interest in this problem should result in a large volume of information being available within the next year or two.

Wood in Automobile Bodies

WOOD is a resilient material that absorbs shocks and sounds. Examples of uses in which this quality is of value are handles of hammers and axes, baseball bats and floors. Motorists look for quietness in the cars that they buy, and the body should absorb rather than amplify the sounds and vibrations. Free-wheeling has made us more conscious of rumble and road noises by intermittently reducing the engine noise, and the resilience and absorptive properties of wood contribute to reducing the noises.

Resilience is probably the reason for wood being less subject to fatigue than are metals. A wooden body frame well designed and made of properly selected wood is tough; it will withstand much punishment without injury.

Many improvements have been made during the last few years in the manufacture and use of wood for bodies. One of these is the more scientific design of braces and reinforcements. Such braces must not be designed to be too rigid for the flexibility needed in the whole assembly, or it will impose unnecessary stresses. We know that the easiest way to break a piece of wood or any other material is to clamp one end of it in a vise.

The constantly accumulating fund of data regarding the properties and uses of the various kinds of lumber that are available in profusion in the South is of great assistance to designers and woodworkers. This enables engineers to utilize the different species for the various parts for which they are most suitable and avoid the dissatisfaction that has occurred occasionally because of the substitution of unsuitable Southern woods for the more expensive Northern species.

Appreciation and acceptance of the fact that lumber, after being properly kiln dried, can be machined accurately to size and maintain its dimensions for assembly at factories far from the mill where it was fabricated have greatly decreased the cost of finished woodwork.

Use of glued-up lumber for many parts requiring a thickness of 2 in. or more, as well as for thinner veneer, results in economy and often in greater strength. The low-cost sills and other frame members now being made of glued-up oak probably are equal to any that have ever been used in the body industry. Progress has been made in the development of water-resisting glue for parts that require it.

Finger-jointing equipment that has been in use for several years has been of advantage in making the contours of bodies and in locating joints so that the grain will be as straight and the members as strong as possible. Inexpensive wood can be used for a considerable portion of a part and stronger wood inserted at any point where the stresses are particularly severe.

An improvement of far-reaching importance which is not yet fully appreciated or widely used is the chemical treatment of wood to prevent decay. This is not done by attempting to seal the wood against moisture but by impregnating it with a chemical that is poisonous to the fungus or vegetable growth causing decay. I believe that the life of wooden members of bodies can be materially lengthened by a treatment which costs less than ½ per cent of the cost of the body.

Research and development affecting the design and manufacture of wood is proceeding more rapidly than ever before. They can be counted upon to give us better bodies and perhaps cheaper ones from year to year. To foretell the future trend accurately is impossible but a reasonable conclusion is that the qualities of wood which have justified its use for so many centuries will continue to be of value to future builders of automobile bodies.—From a paper read at a meeting of the Body Division of the Detroit Section by William L. Hoge, M.S.A.E., president of Mengel Body Co., Louisville, Ky.

Frame Design and Front-End Stability

Semi-Annual Meeting Paper

By Clyde R. Paton[1]

EXPERIMENTAL work done to ascertain the influence of frame and body structures upon front-end stability of the automobile is described by the author and definite means of preventing the phenomena of wheel wabble, shimmy and vibratory movements of the radiator, head-lamps and fenders are set forth.

Early investigation showed that the problem involved not only the unsprung portions of the car but also the structural arrangement of the frame and the body.

Chassis-dynamometer tests revealed a nodal point of zero torsional vibration approximately at a plane through the front seat but varying with different cars and body types, the forward portion of the chassis vibrating torsionally about the longitudinal axis in opposite phase to the rear portion. Experiments rather conclusively proved that damping is needed in the frame and body. Many tests with stiffened frames gave disappointing results on the road as compared with laboratory tests, and it became apparent that the frame and body must be treated as a single structure. A more rigid method of bolting the body to the frame, bracing and reinforcing to increase the body rigidity and securing the radiator more rigidly gave greater front-end stability.

Addition of weight at the front, remote from the neutral axis of the frame, as by placing spring-mounted weights in the ends of the front bumper, proved very effective, the weights being given a frequency substantially the same as that of the torsional vibration of the chassis.

Spare wheels and tires mounted in the front-fender wells, and unsprung weight well out from the center of the axles, have a very beneficial effect, and the author believes that we have gone too far with many cars in reducing the unsprung weight.

Open and convertible cars call for different treatment than is most effective with sedan models, which are stiffer in the mid-section.

ENGINEERS have been faced for a number of years with the problem of suppressing shimmy and the associated vibratory front-end movements of automobiles when driven at high speed or over irregular road surfaces. This has been a general problem and has been particularly in the foreground since the advent of balloon tires, four-wheel brakes and rubber engine-mountings. Much experimental work has been directed toward the solution of this problem and considerable has been accomplished. When first encountered, this phenomenon was regarded as a disturbance involving movements of the front axle and wheels, and most attention was directed toward suppressing these movements. The so-called shimmy or axle tramp encountered at high speeds was thought by many to be a disturbance that was more or less distinct from the vibratory movement of fenders, lamps and radiators. Only recently has the relationship between high-speed shimmy and front-end vibration on rough roads been recognized.

This paper will deal largely with the influence of frame and body structures upon the stability of the automobile when operated under conditions tending to induce shimmy or front-end movement. A summary of the results obtained from a series of experiments relating to the subject, which have extended over the last several years, is presented. I purpose to present experimental evidence that shimmy and front-end shake are not a local front-end disturbance but involve the entire chassis and body; that the desired front-end result usually cannot be obtained by frame-design improvements alone; and that improvements in the rigidity of the frame do not bear a direct relationship to the over-all result obtained with the complete car on the road. I also hope to show that the disturbance commonly known as shimmy or front-end vibration on rough roads is substantially a torsional vibration of the entire chassis about a longitudinal axis and that this vibration cannot, in many instances, be sufficiently corrected by even major structural reinforcements of the frame.

I shall review some of the attempts that have been made to improve the front-end result and offer some suggestions as to treatments which experience has shown to be most helpful. The complete solution has by no means been reached, particularly in the case of the larger, heavier cars. I shall review the fundamental difference in the behavior of cars equipped with the conventional sedan body and those equipped with the open or convertible-type bodies and show that corrective treatment which is found helpful for the conventional sedan type usually does not accomplish a corresponding correction on cars with open or convertible-type bodies. By such evidence my hope is to present a general picture, as I now understand it, that may lead to the development of further corrective means much needed at present, particularly by all open or convertible cars.

Why Some Early Results Were Inconsistent

In spite of a rather thorough and careful investigation of detail factors affecting front-end stability, the fact became apparent during the early experiments that this general problem not only involved the front-end unsprung structure of the car, but also was very definitely related to the structural arrangement of the frame and the body. It was noted that certain cars, practically identical in the matter of frame and body design, behaved in an altogether different manner when driven under identical conditions on the same roads. Certain models in the same line were very much better than others, open cars performed much differently than closed cars, and cars with one type of body structure were different from those having another type of body structure on the same chassis. These apparently inconsistent results have been partly explained by further study. Therefore I shall review some of the experimental results obtained and point out some of the discrepancies between static-deflection tests made in the laboratory and the effect of experimental changes on the road result.

Front-End Phenomena Defined

For the purposes of this paper, it seems desirable to define some of the terms commonly used in a discussion of this subject. The terms shimmy, tramp, wheel

[1] M.S.A.E.—Chief engineer, Packard Motor Car Co., Detroit.

wabble and front-end shake have frequently been used synonymously, but I desire to differentiate between these several terms as follows:

Shimmy.—The disturbance occurring usually at the higher road speeds, induced either by unbalance of the front wheels or by an irregular road surface, which results in a combination usually of wheel wabble, axle tramp and a torsional vibration of the entire chassis. This disturbance is more evident at the front end of the car and often produces severe vibration of the radiator, lamps and fenders, referred to as front-end vibration.

Tramp.—Vibratory movement of the front axle and wheels in a substantially transverse vertical plane, consisting of an oscillatory rotation of the front-axle assembly substantially about a center generally not on the axle but usually above it.

Wheel Wabble.—Periodic rotation of the wheels about the steering-knuckle pivot.

Unless otherwise indicated, I refer throughout the discussion to the conditions encountered with the conventional front-engined rear-drive sedan.

To further qualify my definitions, I should like to state my belief that pure wheel wabble, without some accompanying tramp, does not occur with present conventional front suspensions. My observation is that wabble is always accompanied by some tramp and, conversely, that tramp is always accompanied by some wheel wabble. We all have seen cases of low-speed wabble which were relatively free from tramp. Gyroscopic forces, of greater magnitude at high speeds, account for these two movements being interrelated and always associated in present conventional axle designs. To examine the character of these movements as they occur on the road is not easy, but fortunately it is possible to produce similar conditions on a chassis dynamometer and to operate a car on drums for a more detailed study than could be accomplished by road observations.

By mounting a car on the chassis dynamometer and producing a condition of shimmy by unbalancing the wheels, the character of the wheel and axle movement can readily be observed. Fig. 1 is a typical diagram of the movement at the wheel center as obtained on the dynamometer by placing a pencil at the center of the hub, arranged to trace the wheel path on a stationary card. The relative magnitude of the vertical and horizontal components of this movement are indicated by vertical and horizontal tangents. This diagram shows a normal condition encountered on a car at the critical shimmy speed, which in this case was approximately 65 m.p.h. The car was a typical 4000-lb. five-passenger sedan of 120-in. wheelbase, having one front tire unbalanced by 50 oz.-in.

Observation of a car when mounted on the chassis dynamometer also showed that considerable movement of the front end of the frame occurred whenever axle tramp was produced. At first this was thought to be only a local condition at the front end of the frame, but later we discovered that the disturbance occurred throughout the length of the chassis. A powerful stroboscope revealed that when a shimmy was produced, by either unbalancing the front wheels and operating the car in the critical speed range or placing blocks on the chassis-dynamometer rolls, the disturbance that was occurring was actually a torsional vibration of the frame and body structure about a longitudinal axis which passed approximately through the neutral axis of the frame. The entire front portion of the car appeared to be vibrating torsionally about this fixed axis. Fig. 2 shows roughly the location of this axis and the character and magnitude of this vibration.

A Nodal Point of Torsional Vibration

A nodal point of zero torsional vibration occurred approximately in a plane through the front seat. This point would vary with different cars and different types of body, but for the conventional sedan it usually is approximately in the position shown in the figure. Examination with the stroboscope showed that all portions

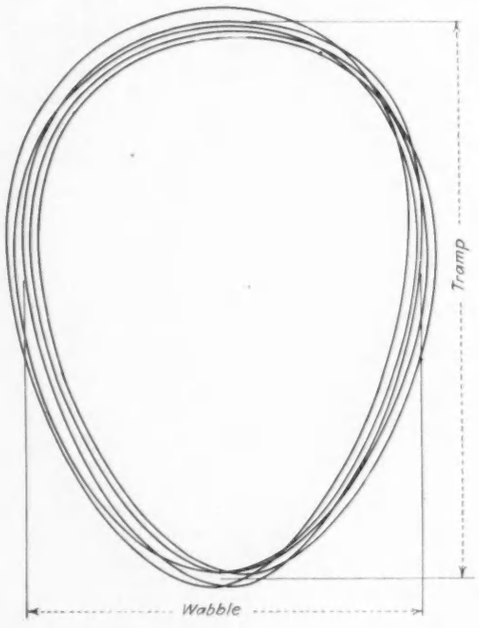


FIG. 1—DIAGRAM OF MOVEMENT OF FRONT-WHEEL CENTER AT CRITICAL SHIMMY SPEED

Obtained on Chassis Dynamometer with Pencil Placed at Center of Hub. Relative Magnitude of Vertical and Horizontal Components Is Indicated as Tramp and Wabble

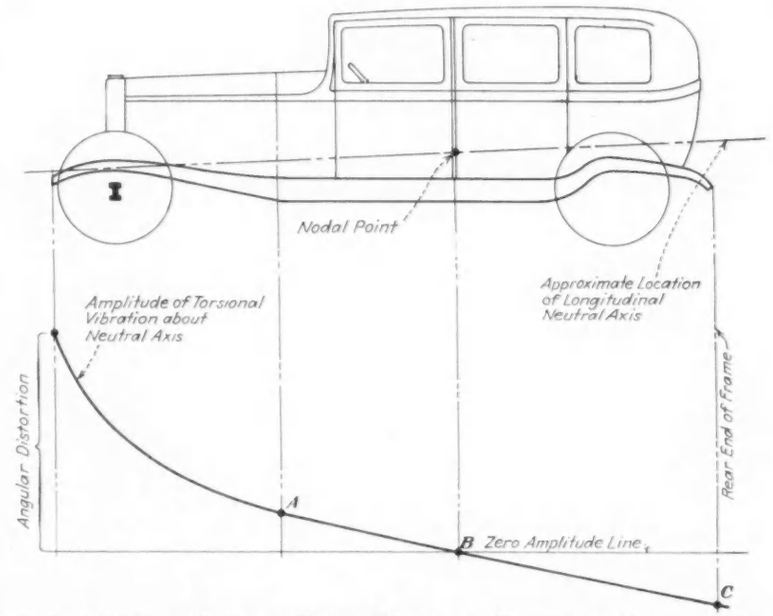


FIG. 2—DIAGRAM SHOWING NODAL POINT OF TORSIONAL VIBRATION AND AMPLITUDE OF VIBRATION ABOUT NEUTRAL AXIS OF A CONVENTIONAL SEDAN

of the car forward of this nodal point vibrate torsionally approximately about the longitudinal axis and always opposite in phase to portions rearward of this point; for example, if the front end is distorted in a clockwise sense, the rear end at the same instant is distorted in a counterclockwise sense. The general magnitude of the angular displacement was found to be much less in the region of the body and to increase rapidly forward of the dash, having the maximum amplitude at the front end of the frame. The lower portion of the diagram shows roughly the general character and relative magnitude of this longitudinal torsional vibration.

This discovery presented a new angle of the shimmy problem. We apparently had discovered the structure of the frame and body to be a relatively undamped torsional spring which, when excited by tramp of the front axle through reaction forces transmitted by the front springs, was free to vibrate torsionally, with very disturbing impressions to passengers.

In the control of shimmy, the fact has been clearly established that balance of the tire and wheel equipment is of utmost importance; also that balance of the tire alone must be held within very close limits and that it is not possible fully to correct for an unbalanced tire by correspondingly unbalancing the wheel. Great improvements have been made in recent years in reducing the unbalance of tires and tubes, but had not the so-called shimmy shackle been evolved and rather generally adopted manufacturers would have been obliged to go much farther in the reduction of tire unbalance than has so far been necessary.

Damping Decreases Tendency To Shimmy

In the case of shimmy, as in other types of vibration, frictional damping has been applied with beneficial results. The importance of spring interleaf friction, adequate shock-absorber equipment and friction in steering connections, as damping forces normally opposing the tendency to shimmy, has been fairly well established. Correctly calibrated shock-absorber equipment had been shown to be very useful in the suppression of shimmy. It was also known that shock-absorbers must be rigidly mounted, as flexibility in their mounting causes poor results. Six-ply tires were found to be superior to four-ply; plain spindle bearings were better than anti-friction bearings; and increasing the number of leaves of the front springs or removing the lubricant from the springs was found to be beneficial.

In every instance in which damping was increased, the shimmy tendency decreased. However, some cars still shimmied. The test results had all indicated that either we had not been able to increase frictional damping sufficiently or that somewhere in the system an elastic structure existed to which adequate damping had not been applied. Later experiments have proved rather conclusively that the structure in which additional damping was needed was the frame and body. Apparently the frame and body, considered as a torsional elastic structure, were in most cases so lacking in hysteresis properties as to be readily excited by the tramping movement of the axle. This hypothesis partly explained differences that previously had been observed between different types of body structure.

The frame itself, although of rigid design, is an elastic structure that is relatively difficult to damp. The body may or may not incorporate a structure providing the necessary damping. Body structures of the wood-frame or composite type showed, in general, less tendency to shimmy than bodies of the all-steel type which, although possibly more rigid than the composite type, are elastic in character and provide relatively little damping against torsional vibration.

It was logical, in view of these observations, to investigate the torsional rigidity of cars, with the thought that frames might be reinforced sufficiently to provide the necessary torsional stiffness. Body engineers argued that economy dictated the elimination of these stresses from the body structure; that frames should be capable of providing the necessary rigidity; that it was necessary to mount bodies of various types, including the open and convertible types, upon the same chassis; and that, if possible, the frames should be sufficiently rigid to permit the use of flexible attaching means between the frame and the body sills. If this could be done, considerable weight could be taken out of the body and a satisfactory result still be obtained with greatly improved body durability. They argued that the structure required in the body to assure safety to passengers was much less than that required to assure satisfactory durability of bodies mounted on relatively flexible frames.

TABLE 1—DEFLECTION RATES OF THREE TEST CARS

	Deflection Rate, Lb. per In.		
Car	*A*	*B*	*C*
Frame Alone	30	80	240
Frame and Engine	120	480	315
Frame and Body	300	500	500
Complete Car	325	910	580

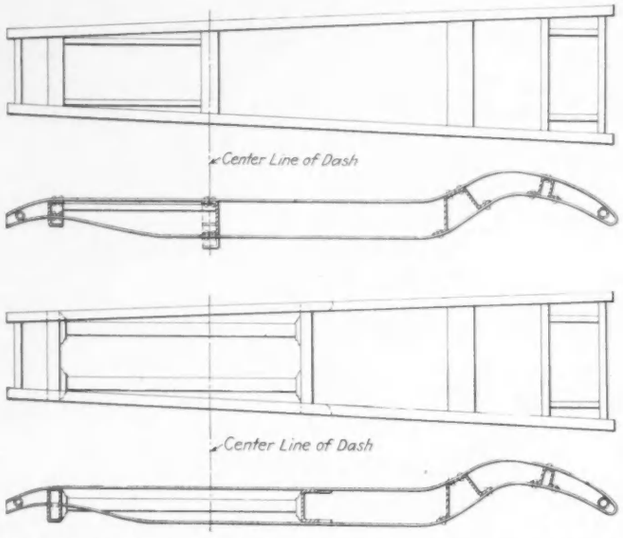
About this time the use of rubber and other flexible means for mounting engines became of interest. General reduction of the noise level, made necessary by improved roads and high road speeds, made it essential that engines be flexibly mounted on the frame to obtain the greater degree of silence demanded. This development further complicated the problem. In all cases the use of flexible engine-mountings greatly impaired the front-end result and increased the demands upon the body designer to provide structures that would be durable and would not develop squeaks and rattles.

Car Torsional Rigidity Investigated

In approaching the problem of providing the necessary frame rigidity, a survey was made of a number of cars to determine the torsional rigidity or the elastic rate of the various components contributing to the total over-all result. A method of test that naturally suggested itself was to mount the complete car at the four corners of the frame, three of the mounting points being pivotally fixed and the frame deflected at the fourth point, measurements being taken of the force required to produce a given torsional deflection, and in this way obtaining a figure representing the torsional elastic rate of the complete structure. Thus the rigidity of the frame alone, of the combination of frame and body and of the combination of frame, body and engine could be separately evaluated. A number of different cars were examined in this manner and considerable variation in total rigidity and in rigidity of the components was found. Frames of different cars, when considered alone, varied from 25 lb. to several hundred pounds per inch of deflection. A surprising variation in body rigidity, when taken alone, was also found, as shown in Table 1.

Car *A* was a typical 127-in.-wheelbase seven-passenger sedan. Fig. 3 shows the general design of the frame. The engine was rigidly mounted on a relatively flexible subframe. The frame alone had a torsional rate of only 30 lb. per in.; addition of the engine increased this to 120 lb. per in., and removing the engine and installing the body brought the rigidity up to 300 lb. per in. The complete car had a rate of 325 lb. per in. of deflection.

Car *B* was of another make but of the same wheelbase and incorporated a rigid engine-mounting anchored directly to the frame at four points. This frame was, in general, of practically the same design as that of Car *A* but had channel cross-members of heavier design which were substantially gusseted to the side rails. The frame alone showed a rigidity of 80 lb.; the frame and engine,



FRAMES USED IN TESTS TO DETERMINE EFFECT OF FRAME RIGIDITY

Fig. 3 (Top)—General Design of Frame of Car *A*, in Which the Engine Was Rigidly Mounted on a Relatively Flexible Subframe. This Frame Had a Torsional Rate of only 30 Lb. per In. of Reflection

Fig. 4 (Below)—Experimental Frame of Car *C*, in Table 1, Substantially the Same as Frame *A* but Having Extremely Rigid Channel Cross-Members under Radiator and behind Transmission, Connected by Heavy Longitudinal Tubes. Its Torsional Rate Was 240 Lb. per In.

480 lb.; the frame and body without the engine, 500 lb.; and the complete car, 910 lb. per in. of deflection.

Car *A* gave a relatively unsatisfactory front-end road result, being very susceptible to shimmy and showing excessive movement of the fenders, lamps and radiator on rough roads; whereas Car *B* gave an extremely satisfactory road result, showing practically no shimmy and no front-end movement. In Car *A* the major part of the rigidity was contributed by the body; and, although the engine and frame combination showed a rate 90 lb. greater than that of the frame alone, the complete car was only 25 lb. more rigid than the frame and body alone. In the case of Car *B*, the frame and engine together showed 400 lb. greater rigidity than the frame alone. Car *B* had both a more effective engine mounting and a more rigid body, and, complete, was almost three times as rigid as Car *A*. Further tests on Car *A* showed that the deflection was occurring largely in the portion of the chassis extending forward from the No. 2 body bolt, and it was reasoned that a corresponding major increase in the rigidity of the frame structure in this forward portion of the chassis should provide a considerable increase in rigidity for the complete car and give the desired degree of increased rigidity.

Road Tests of Rigid Frames Disappointing

A number of experimental frames were built and tested in the laboratory for torsional rigidity. Fig. 4 shows the construction of the outstanding example of these experimental frames. This frame, designated *C* in Table 1 and built into a car exactly the same in other respects as Car *A*, was substantially the same as Frame *A* except that an extremely rigid channel cross-member was installed under the radiator and a second similar member behind the transmission, while heavy tubular longitudinal members were incorporated between them. These tubes were securely anchored to the channels and served as a subframe for the engine. The tubes were 3 in. in diameter and of ⅛-in. wall thickness. This frame, when tested alone, increased the rigidity of Frame *A* from the original 30 lb. per in. to a rate of 240 lb. per in. for Frame *C*, approximately eight times as rigid. Addition of the engine to Frame *C* brought the rigidity up to 315 lb. per in. Rigidity of the frame and body alone was increased from 300 lb. to 500 lb., and that of the complete car from 325 lb. to 580 lb.

This result, obtained in the laboratory, began to look encouraging; however, Cars *A*, *B* and *C* were then tested on the road by a committee whose members were unanimous in the conclusion that no appreciable reduction in tendency to shimmy or improvement in front-end stability had been accomplished and that the ride was more harsh and jolty, presumably because of the increased front-end rigidity.

A large number of other experimental frames also were tried. Side-rail thickness was increased to 3/16 in., with resultant increased structural rigidity but no noticeable improvement on the road. Double-channel side rails extending from the front end of the frame to the mid-point of the body, although increasing the rigidity, as determined by static tests, gave no encouragement on the road. Frames incorporating longitudinal and transverse tubes were tried without success. Side rails of box section and of D-section, shown in Fig. 5, were carefully tested in the laboratory and tried on the road. Heavy tubular and box-section cross-members were built up and tested. Frames were braced and trussed in various ways. Side rails of greater depth showed only minor improvement.

Meanwhile, to avoid the expense of building complete frames, a very instructive series of experiments were conducted with cardboard models which made possible a quick qualitative comparison of various frame details. The most promising models were incorporated in complete frames and tested for rigidity. Several of these were built into cars but, when tested on the road, were found either to give only minor improvement or, in many cases, a less satisfactory result than had been obtained with the original design. Major reinforcements of the frame in the portion occupied by the body did show an improvement in body life and greater freedom from the development of squeaks and rattles when tested for endurance, but, as for improving the general road result and obtaining greater front-end stability and freedom from shimmy, the results were very disappointing.

Frame and Body Must Be Regarded as a Unit

The conclusion, therefore, became forcefully apparent that attempts to solve the problem by frame changes alone were rather hopeless and that it would be necessary to approach the problem of the combined frame and body as a single structure. A few rough calculations indicated that it was wholly unreasonable to expect to develop, in a frame structure having a few inches of depth, the order of rigidity apparently necessary and which could be obtained in the frame and body considered as a structural unit having a depth of several feet. The rigidity which could be obtained in even a frame of complete box section and of unreasonably heavy-gage stock was computed to be only approximately one-quarter of that obtainable in a structure of the magnitude of the conventional body, figured as a box and computed on the basis of conventional body-panel gages. A further comparison of Cars *A* and *B* showed that Body *B* was of virtually the same total weight as Body *A* but apparently was of considerably greater rigidity. This prompted a study of body structure and the means of attaching the body to the frame.

Examination of the conventional methods of body attachment showed much to be desired, and therefore experiments were directed toward improving the rigidity of the body-to-frame attachments. No difficulty was

experienced in improving upon the design previously employed by making detail modifications of the frame brackets and by more effective location and an increase in the number of body-holddown bolts. Details of the original and modified attachments are shown by Fig. 6. These changes resulted in a noticeable increase in the over-all rigidity and in a distinct improvement when the car was tested on the road.

The question as to the effect of these changes upon body life immediately arose. Therefore two cars were prepared and tested for life under identical conditions over all kinds of road. The first car was standard in every respect, with body-to-frame attachments of conventional design having holddown bolts passing through the wood sills and only the upper flange of the frame, the bolts being located at points in general midway between the body pillars. On the second car, substantial reinforcing brackets were attached to the frame, the bolts being located at the base of the pillars and anchored securely to both flanges of the frame. Where possible, extra bolts were installed to provide a more secure attachment between the body structure and the frame side-rails. The road result was surprisingly better with the second car than with the first.

These two cars were operated together for approximately 20,000 miles under severe road conditions, and at the end of the test the result, although not as satisfactory in either case as at the start, showed that the second car, having the improved body-to-frame attachment, had undergone less body depreciation, had required less servicing during the test, and was in much better condition than the car having the original, less substantial method of body mounting. The standard car had undergone a surprising depreciation as a result of this rather severe road test. It had been necessary during the test to twice service the door locks and dovetails. At the end of the test the door hinges were in poor condition; the dash had failed on both sides, necessitating repairs during the interval; and the body had a

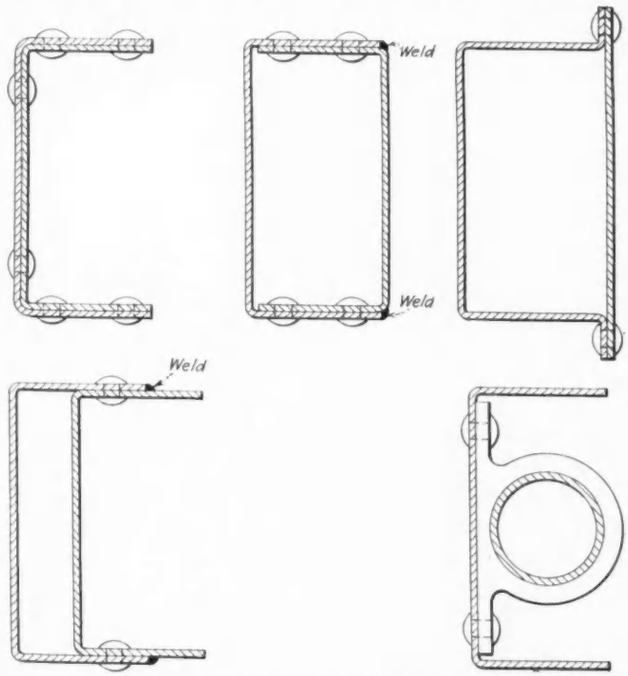


FIG. 5—CROSS-SECTIONS OF EXPERIMENTAL REINFORCED FRAME SIDE-RAILS

Such Construction Stiffened the Frames and Gave Greater Body Life and Freedom from Squeaks and Rattles but the Increased Rigidity Did not Noticeably Improve Front-End Stability

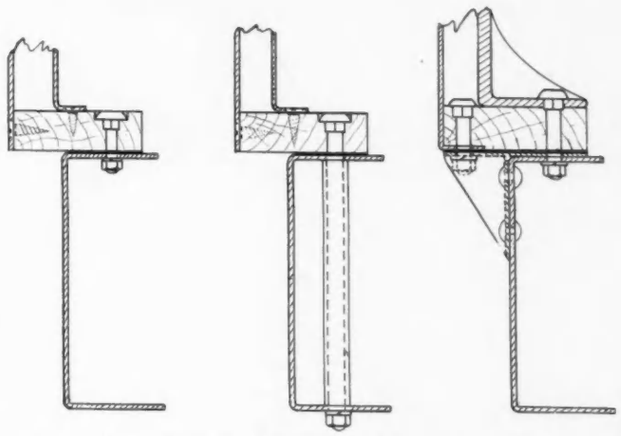


FIG. 6—ORIGINAL AND MODIFIED METHODS OF ATTACHING BODY TO FRAME

Methods Shown at Center and Right Noticeably Increased Over-All Rigidity of the Car and Distinctly Improved Stability on the Road

large number of squeaks and rattles and was ready for the scrap heap. The general road result was very unsatisfactory as regards both shimmy and front-end movement. The special car, on the contrary, came through the same test with practically no servicing required and was in a relatively satisfactory condition. Road observations on this car at the end of the test, although showing some depreciation, were relatively satisfactory as compared with the new-car result.

To determine how much of this depreciation of the standard car had occurred in the frame or other chassis units, a new production body was installed on this 20,000-mile chassis. So far as could be judged by laboratory tests or road observations, practically the entire depreciation had occurred in the body.

To ascertain what portions of the body structure had suffered the greatest depreciation, the cars were returned to the laboratory for further tests, which revealed that a large part of the decreased rigidity could be traced to the roof panel. Diagonal roof-bracing extending from the windshield pillars to the rear-quarter structure increased the rigidity approximately 30 per cent and brought the total rigidity back to within approximately 10 per cent of the result when new. Substantially the same improvement was obtained by replacing the "chicken-wire" roof covering, indicating that a large part of the roof depreciation had been due to a permanent set or stretch in the wire. Further experiments conducted by installing diagonal bracing in various positions showed that the dash and cowl structure and side panels had suffered the remainder of the depreciation. Considerably better results than those obtained with the original new standard body were obtained by reinforcing the dash and side panels. The importance of a substantial cowl structure and of rigid attachments between pillars, body sills and roof rails, as well as rigidity of the structure in the rear quarters and back, was clearly demonstrated.

Problem Presented by Flexible Engine-Mounting

Having incorporated all feasible detail changes in the body structure, the front-end result on the road was considerably improved and would have been regarded as very satisfactory had it been possible to continue with a rigid engine-mounting. Even with flexible engine-mountings at the rear end of the powerplant, the front-end result was fairly satisfactory. However, to obtain the reduction in noise level demanded, powerplants had

to be fully insulated from the frame, hence other and further means of increasing front-end stability must be found. We had obtained rather definite proof that the body must be considered as the major structural unit and that the frame, rather than being considered as something upon which to mount the body, should be regarded as an extension of the body on which to mount the other parts of the car.

Major reinforcements of the front end of the frame were again considered in the hope that, having accomplished considerable improvement in the structure to the rear of the dash, a satisfactory front-end result might be obtained by extending a frame reinforcement sufficiently to the rear and tying the frame substantially to the body structure. Some improvement was obtained by the use of rather elaborate and costly frame designs. The X-type cross-member, incorporated midway of the body and extending forward to the radiator support as an additional side-rail reinforcement, did show some improvement; but excessive reinforcement of the frame extending forward from the dash was found to impair the general result, as in several cases it increased the harshness of the ride to an undesirable extent. Measurements obtained in the laboratory showed that cars with these major front-end frame reinforcements considerably increased the torsional rigidity, yet road observations were relatively disappointing. When engines were fully rubber-mounted, even frames having the greatest possible amount of front-end reinforcement obtainable with combinations of riveted and welded structures did not give the degree of front-end stability that had previously been obtained with cars having engines rigidly mounted on frames of less elaborate design and which did not exhibit acceptable road stability.

Laboratory tests had proved that an engine mounted on rubber supports at the rear but rigidly attached at the forward end contributed relatively little to front-end rigidity. Yet the rigid mounting of the engine at the front end of the frame was shown by road tests to give a much better result than when mounted flexibly at all points. Apparently, in mounting the engine on rubber, we had lost more than could be accounted for by the rigidity of the engine itself. We were forced to admit that no definite relationship existed between the torsional rigidity as obtained by the means we had employed in the laboratory and the result obtainable on the road. The fact became evident that the development must be continued on the basis of road results rather than on static tests for torsional rigidity.

Instrument for Measuring Vibration in Road Tests

Road tests were inconclusive because of the difference of opinion among individual observers, hence some instrument for measuring vibration of the various portions of the car on the road was required. A number of different devices were tried, the most satisfactory one found being the instrument shown in Fig. 7. This is an adaptation of the ordinary pedometer and consists of a pivoted weight which, through the action of a fine-tooth ratchet operating a hand on a suitable dial, gives a single reading roughly proportional to the number and amplitude of the movements being measured. When mounted on any portion of the car, it can be used for measuring either vertical or transverse vibration. When used as a transverse instrument, the balancing hair-spring is removed.

These instruments proved invaluable in making road observations, as they eliminated the differences of opinion of individual observers and made possible a determination of differences of smaller magnitude than those possible of evaluation by observation only. Pedometers attached to the radiator cap, head-lamps, fenders, windshield or other parts of the car were very useful in

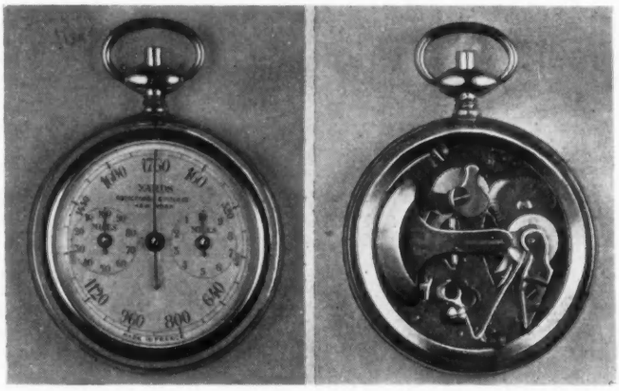


FIG. 7—MODIFIED PEDOMETER FOUND MOST SATISFACTORY FOR MEASURING NUMBER AND AMPLITUDE OF VERTICAL AND TRANSVERSE VIBRATIONS

measuring on the road the influence of various factors. The method of testing consisted of driving the car at the critical driving speed over a given road surface, under conditions established as standard for comparison. Surprisingly consistent results were obtainable.

Effect of Side Weights at the Front

Road tests were continued with the aid of the pedometer equipment to determine why the rigid engine-mounting at the front end of the frame contributed more to the improvement of front-end stability than could be explained by its influence upon rigidity. The mass of the engine, rigidly anchored to the forward end of the frame, was suspected of contributing the beneficial influence. To prove that this was the governing factor, a construction as shown by Fig. 8 was attached to the forward end of the frame of a car adjacent to the front supports of the full-rubber-mounted engine. This addition of weight remote from the neutral axis of the frame had a very beneficial effect; in fact, a better result could be obtained by attaching sufficient weight than was originally obtained with the rigid engine-mounting.

The tests seemed to prove rather clearly that the stability which had been lost in mounting the engine on rubber was not due so much to rigidity as to the effect of the engine mass on the polar moment of inertia about the neutral axis of the frame. Therefore weights were tried at various places and found most effective; for example, at the ends of the front bumper. Equivalent weights located at the center of the forward end of the frame were much less effective than when installed as far out from the neutral axis as possible. Use of fixed weights of the magnitude required to accomplish the desired result was, of course, impracticable; but, having established the desirability of increasing the polar moment of the front end of the chassis, an obvious second step was to consider means of utilizing to the best advantage the masses already available.

The fact that spare wheels and tires, when mounted in front-fender wells, gave greater front-end stability than when carried at the rear had already been observed. Some further improvement was obtained by placing the battery and tool compartments in the front fenders, thus providing better utilization of these necessary weights. The radiator assembly, if properly utilized, also seemed to be a logical weight to use, particularly as its center of gravity was favorably disposed with reference to the neutral axis of the frame about which the oscillation occurred. Dependence upon means previously used for attaching the radiator to the frame was not satisfactory, as rigid attachment of the radiator to the frame cross-

member could be accomplished only to a certain degree with conventional methods of mounting. Further, a certain amount of insulation was required between the radiator and the frame so as to prolong the life of the radiator structure.

Radiator Weight Utilized for Stabilizing

Fig. 9 shows a construction that was worked out to make full use of the radiator weight in stabilizing the front end. The radiator core is partly insulated from the frame by the use of rubber or other elastic supports. The dash, being relatively free from movement, is used as the foundation to which the radiator shell is anchored by means of brace-rods that prevent lateral movement of the radiator with respect to the dash. We next tied the radiator shell, suitably reinforced, to the frame side-rails through the intermediate connection of lamp brackets and front-fender brackets. This construction accomplished two objects: It increased the polar moment of the front end of the frame by the semi-rigid incorporation of the radiator mass as an integral part, and connecting the frame to the radiator shell assured that the slight radiator movement permitted would be substantially in phase with the movement of the frame.

With the older conventional types of mounting, the radiator movement was frequently opposite in phase to the movement of the frame. By securely tying the radiator to the dash and in turn the frame to the radiator, some increased restraint of the frame against torsional movement is also obtained. To some degree this treatment may be considered as camouflage for the slight movement of the frame which is still allowed; however, this movement is relatively small and not without its advantages, as already explained. The stresses in the connecting parts are relatively low, as shown by proving-ground and service experience.

In the case of all except the larger and heavier cars, this expedient has been found to give a satisfactory front-end result, and several modifications of this general treatment may be found on current production models of cars of several makes. In one case the radiator is secured to the dash, the fenders and lamps are secured to the radiator, and both radiator and fenders are flexibly attached to the frame. In this installation the frame is permitted to have considerable torsional freedom that is independent of the fenders and radiator. Radiator mass, in this instance, is effective in suppressing movement of the fenders and lamps but is not effective in stabilizing the forward end of the frame. From the driver's seat, this construction gives a very satisfactory appearance.

Another modification consists of a secure attachment of the radiator to the dash, with an attachment extending from the radiator to the fenders incorporating a spring-tensioned frictional connection. Another maker reinforces the front end of the frame torsionally by a construction consisting of V-type radiator supports having attached at the forward point of convergence a second V-type member extending downward to an attachment at the frame side-rails. This construction provides increased rigidity for the forward end of the frame by effectively extending the body structure forward to a plane immediately behind the radiator and does not utilize the radiator shell as a part of the structure.

Spring-Mounted Weights Used in Bumper

Having recognized chassis torsional vibration about a longitudinal axis as the basic problem involved in obtaining front-end stability, it seemed logical that, in cases where additional correction was necessary, some means of damping or otherwise preventing this vibration might be found.

I have referred to the result obtained by the use of fixed weight, as at the ends of the front bumper. In the case of a 4000-lb. car, we found that an extremely satisfactory front-end result could be obtained by the use of a fixed weight of approximately 30 lb. mounted at each end of the bumper; and the same result could be obtained with much smaller weights by mounting the weights on the ends of a transverse spring and so proportioning the elastic rate of the spring and the weights as to give the combination a natural frequency substantially the same as that of the chassis torsional vibration. This crude device was tried on a number of different cars, with uniformly encouraging results.

We next tried utilizing the bumper as all or part of the weight required, pivoting the bumper at its mid-

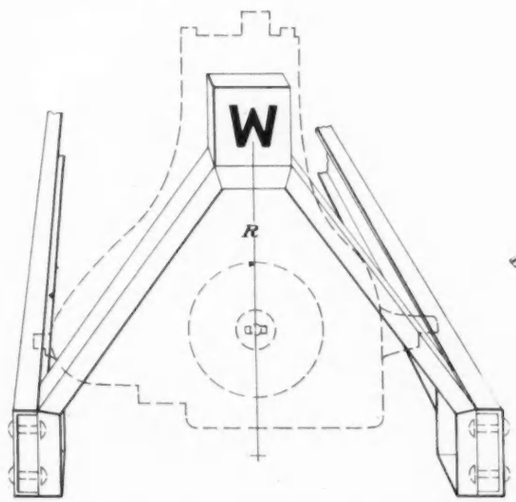


FIG. 8—MASS ANCHORED TO FRONT END OF FRAME FOR TESTS

Construction Used To Demonstrate that Mass of the Engine when Rigidly Attached at Front End of Frame Is More Important than Rigidity of the Engine in Improving Front-End Stability

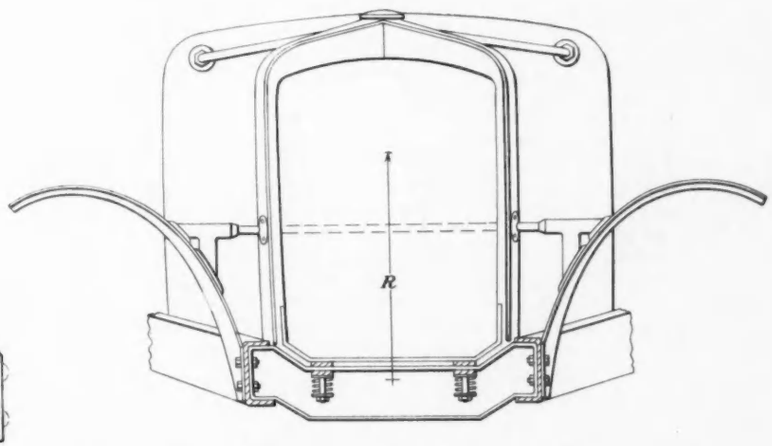


FIG. 9—METHOD OF USING WEIGHT OF RADIATOR FOR STABILIZING FRONT END

The Radiator Core Is Insulated from the Frame by Elastic Supports. The Radiator Shell Is Reinforced, Anchored to the Dash, and Tied to the Frame Side-Rails through Lamp and Fender Brackets

point on a fixed support and utilizing the back bar as a spring to obtain the required frequency. This construction gave a relatively satisfactory improvement and was modified to obtain the desired effectiveness by the addition of small weights at the outer ends. The device seemed promising from all angles except that, for relatively small frame movements occurring on even fairly smooth pavements, the bumper assembly moved through an amplitude of possibly 1 in. at its ends and presented a very unsatisfactory appearance.

The next construction tried consisted of a conventional bumper rigidly secured to the frame and the addition of spring-supported weight members which could be more or less concealed from view by the bumper bars. Constructions of this type were developed and found to have a satisfactory spring life.

The final development, shown in Fig. 10, comprised a conventional bumper and mounting with weights slideably mounted on a guide and supported by double coil springs, all placed in the ends of the bumper. The inclosure for the spring-supported weights was made oil tight, allowing splash lubrication of the moving parts. This construction was applied to a 5400-lb. car and has given a most satisfactory account of itself both on the proving ground and in service.

The general range of frequency for all of the cars to which these devices were applied has been found to be in the neighborhood of 500 to 600 cycles per minute. The size of the weight required to obtain the desired improvement depends upon how much front-end movement exists before its application. In general, the heavier the car is, the greater must be the weight to obtain a given result; weights of approximately 11 lb. were found ample to produce the improvement sought in the case of the 5400-lb. car mentioned.

The stabilizer finds its greatest usefulness in the reduction of chassis torsional vibration occurring on rough roads. It reduces the high-speed shimmy tendency, improves the general handling of the car at speed on road surfaces of all kinds and helps the general ride by reducing the torsional vibration frequently noticeable as "quiver." Similar stabilizers have been applied also to the rear end of the chassis and, although not as effective as at the front, have further improved the roadability of cars driven under severe road conditions.

The action of the stabilizer may be roughly illustrated by Fig. 11, in which Curve *A* represents diagrammatically the torsional vibration of the chassis with no stabilizer equipment. An elastically supported weight having substantially the same frequency as the chassis structure and placed so that it is excited by the frame movement will operate substantially opposite in phase to the movement of the frame, as represented by Curve *B*. Addition of the stabilizer reduces the frame movement to that represented by Curve *C*.

Experiment has shown that the spring-supported weight will take on a considerable amplitude of movement for a relatively small frame movement, so it is possible to obtain a neutralizing force of considerable magnitude by the reaction of the tuned weight to oppose the tendency of the frame to vibrate. In practice, some damping is provided which results in a time lag, hence the movement of the tuned weight is not exactly opposite in phase to that of the frame. The damping is obtained by partly filling the stabilizer compartment with oil, which serves the additional object of providing lubrication for the working parts.

The curve in Fig. 12 shows the variation in front-end result obtained by varying the frequency of the stabilizer and measuring road vibration with the aid of pedometers. The frequency was changed in this case by varying the weight; but we found that, if some damping is provided, the adjustment of the stabilizer is not extremely critical; in fact, a single specification generally is satisfactory for various body types on a given chassis model and for the normal variation of tire air pressure encountered in service.

Up to the present time it has been found impractical to construct the frame or body of a heavy car so as to obtain the degree of freedom from front-end movement that is rather easily obtainable on lighter cars of shorter wheelbase. Use of the stabilizer to obtain the desired result on the larger models has been found economical in both cost and weight. That this device might also be utilized with economy on some of the smaller cars, with a considerable saving of weight in both frame and body, is not beyond the range of imagination.

Open and Convertible Cars Less Rigid than Sedans

The condition I have described and shown as existing with conventional sedan bodies is considerably altered with open or convertible types. In the case of the sedan, the torsional-elastic rate is relatively constant from the rear body-bolts forward as far as the dash, then decreases rapidly toward the front end of the frame. In the case of the open type, the rigidity imparted to the frame by the body is neither as great as in the case of the conventional sedan nor as uniform throughout the length of the body. The portion from the rear body-bolt forward to the front seat is, in most cases, only moderately rigid, and the region from the front seat forward to the plane of the windshield is very flexible. The cowl contributes some rigidity, but this is relatively little compared with that imparted by the conventional sedan. Rigidity forward of the dash is practically the same as in the conventional sedan. The structure as a whole may be regarded as similar to the sedan structure except that the portion of the body between the front seat and the windshield has been removed. Doors contribute relatively little rigidity, and even with tops raised the structure is very flexible in this area; therefore the frame must be depended upon almost entirely to contribute the rigidity between the forward and rear portions of the car. The net result is that the front and rear portions are moderately rigid but are connected at the front compartment by a section having relatively low rigidity.

In the conventional sedan the greatest torsional disturbance is evident at the forward end of the frame, while in the open or convertible types it is most evident in the mid-section, which has the lowest rigidity. Whereas front fenders, radiator and lamps of the sedan have the greatest tendency to vibrate, these parts of a car equipped with an open or convertible body are relatively stable and, instead, the windshield, cowl, instrument-board and steering column vibrate to an undesirable degree.

Obviously, the open body should have a much more rigid frame at its mid-section than is required in the case of the sedan, but it is impractical if not impossible to reinforce the frame in this section sufficiently to develop the order of rigidity required to compensate for the relative lack of support provided by the body. Even if this were possible, it probably would not be considered economically feasible to provide a special frame design for the relatively small percentage of open and convertible cars sold. On the other hand, we would hesitate to impose the necessary additional weight and expense upon the conventional sedan, which does not require reinforcement in this region.

Some reduction of the disturbance commonly known as cowl vibration has been attained in the last few years with reinforced-frame design. However, this is still a problem with most manufacturers. It is distinctly different from that of reducing the movement at the front end of the conventional sedan. Many of the corrections that are found helpful on the sedan show relatively little effect upon the open car. Here again the body designer

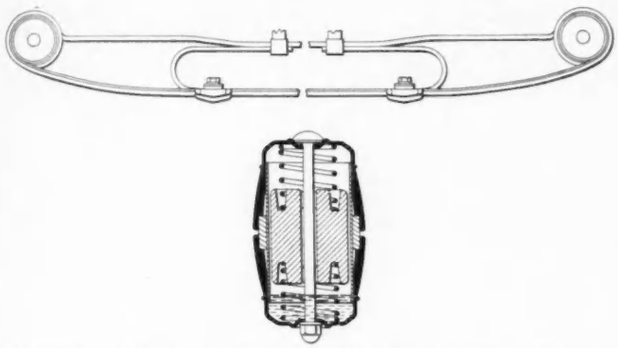


FIG. 10—FRONT BUMPER AND CROSS-SECTION OF TUNED WEIGHT

A Pair of Such Weights Carried in the Ends of the Bumper and Having a Frequency Corresponding to the Natural Frequency of the Chassis Operate in Opposite Phase to Movement of the Frame and Greatly Reduce the Torsional Vibration

has been called upon to so reinforce the structure as to reduce this mid-section movement. Dashes have been reinforced, heavy cast and forged reinforcing members have been employed in the pillars, and some designers have resorted to extra-heavy-gage metal in the cowl. Various types of bracing have been employed but, in general, relatively minor improvements have been attained by this means. I shall not attempt to offer an answer to this problem further than to suggest treatments that might be found helpful.

Road Shocks Should Be Absorbed in Tires

Anything that can be done to reduce the shocks imparted to the frame will improve the stability on both open and closed bodies. Specifying a lower air pressure in tires on open cars has sometimes been found feasible. This can be partly justified by the reduced body weight, and, in cases where high-speed shimmy is not a serious problem, has been found helpful.

The trend toward the new low-pressure tires probably will do as much to solve the open-car problem as any one item. The extent to which road shocks are absorbed by tires is a function of the air pressure and, to some extent, of the construction of the tire carcass. Four-ply tires are noticeably better than the six-ply in their ability to absorb road irregularities.

One factor which I believe has been rather generally overlooked is unsprung weight. Other things being equal, the greater the weight of axle, wheels, brakes and springs is, the less road shock is transmitted to the frame and the more is absorbed by the tires. Here again the distribution of weight of these units is of considerable importance. The most effective weight distribution is that which presents the greatest polar moment about the mid-point of the axle; by this I mean that weight at the wheel hubs is more effective than the same weight at the middle of the axle. To illustrate, if wheels were made very heavy, relatively little road shock would be transmitted through the springs to the frame and most of the road irregularities would be absorbed by the tires. Conversely, if wheels, brakes, axle and springs were very light, road shocks transmitted to the frame would be severe. The amount of mass localized at the outer ends of the axle controls to a considerable extent the division between the tires and the car springs of the work of absorbing shocks from road irregularities.

The recent trend toward wheels of smaller diameter, fitted with drop-center rims, has made our problem more difficult by causing a general reduction of wheel weight without necessarily a corresponding reduction in tire pressure. Fortunately, the advisability of increasing the thermal capacity of brake-drums partly offsets this reduction in weight. I can easily excuse the moderate increase in weight caused by the several types of composite and cast drums.

I believe that the unsprung weight of many cars now on the market has been reduced below that which gives the best all-round result. This has been clearly demonstrated in several cases in which it has been found possible to greatly reduce the general harshness of ride and improve the front-end result by loading with additional weight either the axle at the outer ends or the wheel hubs. If, as now seems to be the trend, low-pressure tires are rather generally adopted, this effect can be obtained in this manner without the use of otherwise unnecessary weight.

How Weight of Parts Can Be Used Effectively

Considerable improvement can sometimes be obtained by reducing spring rates and shock-absorber settings. In some cases it has been found advisable to specify softer springs and lower shock-absorber settings for convertible and open cars than are specified for closed cars. In approaching the open-car problem, we can take advantage of fender-mounted spare-wheel equipment securely anchored to the frame and dash to reduce windshield and cowl movement. A more effective utilization of the powerplant weight might be worked out. One manufacturer has obtained a certain amount of stabilizing effect at the forward end of the frame by placing the single insulated front engine-support relatively high above the neutral axis of the frame, thus making the mass of the front end of the engine effective in increasing the polar moment at the front end of the chassis. This same treatment has been tried in the case of open cars to stabilize the windshield and dash, by providing a means of attachment between the rear end of the cylinder block and the dash. I might appropriately also suggest that the weight of the battery be utilized by locating and mounting this unit so as to obtain some mass stabilizing effect from it.

Additional fixed weight, correctly placed at the mid-section of the frame, rigidly secured to the frame struc-

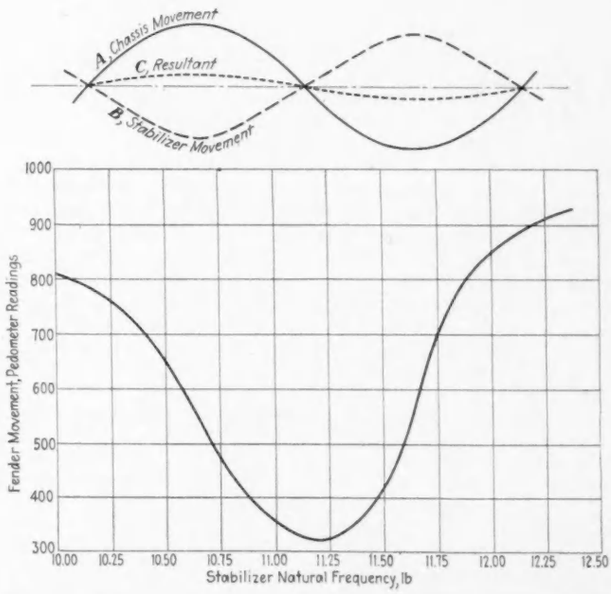


FIG. 11 (TOP)—CURVES ROUGHLY REPRESENTING NEUTRALIZING EFFECT OF BUMPER STABILIZER ON CHASSIS MOVEMENT

FIG. 12 (BELOW)—CURVE SHOWING EFFECT ON FENDER MOVEMENT OF VARYING FREQUENCY OF STABILIZER

ture and located as far as possible from the neutral axis, undoubtedly would be helpful. Likewise, where more refinement is justified than can be obtained by constructions already tried, tuned mass-stabilizers of relatively cheap construction could be employed, concealed either outside the frame side-rails or forward of the instrument panel.

I have shown, I believe, that the problem of chassis stability is not necessarily one of frame rigidity, and have tried to show that repeated attempts to obtain a satisfactory front-end result by frame reinforcement alone, even though carried to extremes, has not produced the result on the road which static tests in the laboratory indicate might be expected. I believe I have also shown that the so-called front-end disturbance commonly known as shimmy or fender vibration is not a localized front-end condition but actually is a torsional vibration of the frame and body structure extending throughout its entire length. The importance of considering the frame and body as a single structure rather than as independent units has been pointed out, and I have suggested that the frame, rather than being regarded as a structure upon which to mount a body, should be considered as an extension of the body upon which the rest of the units are mounted. The point has been made that the apparent discrepancies between results which might have been expected, based on static-deflection tests for rigidity in the laboratory, and actual results obtained on the road could be explained by the influence of mass-inertia effects, and I have shown that polar inertia moments about the neutral axis of the frame are of major importance, particularly at regions having the lowest torsional rigidity. An attempt has been made to show the importance of unsprung weight as affecting the magnitude of the road shocks transmitted to the frame, and several treatments that have been helpful in obtaining stability have been suggested.

I desire to acknowledge the valuable cooperation extended by D. G. Roos and W. S. James, of the Studebaker Corp., in permitting the use in this paper of the test results obtained by the research department of that company.

THE DISCUSSION

C. A. TEA[2]:—The paper by Mr. Paton, covering years of experimental work, represents a stimulating contribution to the problem of automobile stability, particularly with regard to front-end shake. That the work of the Chrysler Corp. on frames has almost exactly paralleled that described by the author is particularly singular.

Mr. Paton may not have gone far enough in his experiments to get the total car stiffness sufficiently high to restrict frame distortion to a negligible amount. In our work we have found, in every case, a critical rigidity, more defined in some cases than in others, above which definite improvement was accomplished.

When the torsional rigidity of the whole car is considerably increased, the need of more flexible springs, particularly front springs, is at once recognized; and, provided such a change can safely be made, a much improved ride is accomplished as well as a material reduction in front-end shake and disturbance of the whole body. Reinforcing the brackets holding the head-lamps and fenders to the frame is sometimes very effective in preventing these parts from shaking. It is possible, though not yet practical for various reasons, to soften the front springs sufficiently, say 40 per cent, on a given car to gain almost complete freedom from body and frame shake as induced by road inequalities, although periodic tramping due to unbalance may even be aggravated at some lower speed. Although increasing the polar moment of inertia of the suspended mass accomplished both reduction in amplitude and frequency of twist, according to the author, the frequency undoubtedly is by far the more important in its effect upon the car and the passengers.

Many Factors Involved in the Problem

The main factors involved in frame and body stability on the road are difficult to state, since we have no relative quantitative data with which to work. What few data we have relate only to particular cars. However, no doubt it is true that rigidity and polar moment of inertia of the whole car, static deflection of the springs, and shock-absorber damping are the principal governing factors in the stability of a particular car and its freedom from shake. Other factors, such as wheelbase, weight distribution, ratio of unsprung to sprung weight, and height of center of gravity, play important rôles in the general analysis of the problem.

My contention always has been that the nodal point in torsional vibration lies near the center of the wheelbase, and I am glad to have this substantiated by Mr. Paton's experimental work. The curve in Fig. 2 of the paper shows one outstanding result, I believe; that is, the definite weakness in the frame side-channels ahead of the dash. Stiffening these torsionally does abet front-end stability, according to our own results.

While low tire pressure aids considerably in the abatement of shake at the lower speeds over rough surfaces by more absorption of energy by the tires, at the higher speeds over uneven paved roads the fender and lamp shake is sometimes worse, because of the vibratory effects of the front-axle assembly. As has been pointed out, a combination of soft tires and heavy unsprung weight might be desirable from the viewpoint of frame and body shake, owing to the smaller displacements of the axle. However, the fact should not be lost sight of that lower unsprung weight in the rear is greatly needed at present to give cars more road-holding ability at high speeds over rough roads.

Knowledge of Torsional-Resistance Requirement Chaotic

The total car rigidity can easily be found by inserting blocks between the axles and frame at two diagonally opposite points, tying down the free rear corner and then dropping the free front wheel over a pit. The amount of deflection of a point on the frame above the axle or at the bumper connection represents the twist under the load carried by one front wheel as weighed on a scale. Rigidity of the car then is roughly obtained by dividing the total load on one front wheel by the deflection.

Although we all have mathematically established methods for designing a frame and body for vertical loads, our knowledge of torsional resistance requirements is so chaotic that a most valuable contribution would be made if someone would develop an empirical mathematical analysis for the predetermination of frame and body rigidity in relation to spring deflection for a car of given wheelbase and weight, assuming the conventional weight distribution.

[2] Research engineer, Chrysler Corp., Detroit.

Chrysler Shake, Rigidity and Fatigue Tests

T. E. M. WHEAT[3]:—The inexpensive modified pedometers shown by Mr. Paton have been very useful in studying vibration. We have used such instruments in working on road shake of all kinds, such as that of radiator, head-lamps and body, and also on engine movements. The standard pedometer is modified by removing the hairspring and as much as possible of the oscillation-controlling devices. The instrument is called a "wabblemeter" for want of a better name and consists of a small, free pendulum enclosed in a watch-like case and operating a recording hand. Test runs are made over a certain road, about 5½ miles long, having a surface of choppy concrete. Tire pressures are carefully checked, and the run is made at a certain speed, usually 60 m.p.h. The "wabblemeters" are mounted on the radiator and lamp bracket or any other part that is being studied and are read before and after the run. I agree with Mr. Paton that the consistency of the results is very convincing.

In our standard rigidity test we do not mount the car frame at the extreme corners, but at the front ends of the springs. One front corner is raised or lowered 1 in., and the force required to thus distort the frame is measured. The reason for mounting the frame in this manner is principally that we thus get a direct check on the center and front part of the frame, disregarding the rear portion. The center and front portions are made relatively much stiffer than the rear portion, because of the flexibility of our engine mounting; therefore we cannot depend upon results obtained from a test that involves the weaker rear kick-up section. Another reason for this test mounting is that it gives a length between supports equal approximately to the wheelbase of the car.

We also use a fatigue test, in which the frame is tested to destruction. The frame is mounted as described and as shown in Fig. 13, but an oscillating device takes the place of the simple distorting force used in the rigidity test. The speed of this oscillating device controls the number of oscillations per minute, and, indirectly, the amplitude of these oscillations. By this means, the frame is shaken to destruction.

This test does not reproduce road conditions and therefore gives no index of the durability of a frame; but such a test is of great value in determining weak spots in any given design, and all weak spots must be eliminated when we start out to save weight.

One result of this test is proof that no torsion occurs in the outside rail adjacent to the X-section of the X-braced frame. Therefore we can design these elements as latticed girders with a number of holes at certain points. Tests show that these holes have no effect on the torsional strength or on fatigue failures.

Frame Stiffness Does Not Cause Breakage

The general impression seems to be that a stiff frame breaks more easily than a limber frame. Not the slightest danger exists of getting frames too stiff. A metallic object breaks because of fatigue effects, and these are a direct function of the stress to which the object is subjected. To break a piece of wire, we bend it first as far as possible in one direction, then as far as possible in the opposite direction. Thus the wire is broken with the minimum number of repetitions. The reason for this is that we are producing a larger stress the further we bend the wire.

A stiff frame will break at the same number of repetitions as a flexible frame, other conditions being equal. Suppose a force of 100 lb. is applied at the corner of a stiff frame and causes a distortion of 0.1 in. Assume

[3] Chassis engineer, Chrysler Corp., Detroit.

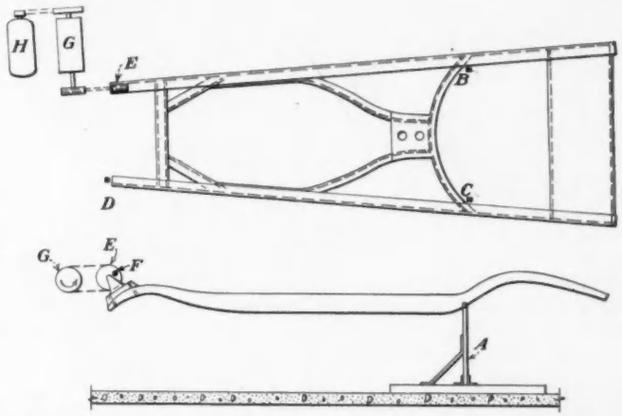


FIG. 13—CHRYSLER SET-UP FOR FRAME FATIGUE TESTS

A, Rack to Which Frame Is Fastened at Points, *B*, *C* and *D*. The Other Front Corner Is Unsupported and Has Mounted on It, as an Oscillating Device, a Flywheel, *E*, Carrying an Eccentric 1-Lb. Weight, *F*. The Flywheel Is Driven by Belt from a Variable Speed-Control, *G*, in Turn Driven by an Electric Motor, *H*

that this produces a stress in the material of 10,000 lb. per sq. in. Now, suppose a similar frame is tested, with the difference that a force of 100 lb. produces a distortion of 1 in. The stress produced in the second frame is exactly the same as that produced in the first, although the deflection is 10 times as great.

To prove this, consider the familiar formula for finding stress from the bending moment. This formula is $f = My/I$, in which f = unit stress, I = moment of inertia of section resisting the bending moment, M = bending moment causing this stress and y = distance from center of gravity of section to extreme fiber of member. The bending moment is that caused by the applied force. In the numerical example given, this force is assumed as 100 lb. In computing the bending moment, this force must be multiplied by the distance to the point at which the stress is to be computed. The stress is the same, no matter how much the distortion.

Some frames are more rigid than others because the metal is used more effectively by good design. One aim of every good designer is to make all the metal in a structure work at about the same unit stress. A weak spot in a frame yields a little, letting the frame deflect more than another frame of better design having no weak spot. When we have found and strengthened all the weak spots, the result invariably is that we can reduce the weight of other parts of the frame. This results in a frame that is both lighter and stronger than we had before.

Frame Rigidity Reduces Front-End Shake

How stiff a frame should be depends on the car design used. For the conditions we have, the following formula gives a good approximation:

$$\text{Frame Factor} = 200 = WL/y \qquad (1)$$

where

L = wheelbase in inches
W = front-wheel load in pounds
y = rigidity in pounds per inch of distortion, determined as stated above.

We have found the figure of 200 to be about right for cars giving an acceptable road performance, using "floating power" and all the other characteristics of Chrysler design. This road performance is determined by using the modified pedometers described, and for our conditions we try to hold this figure constant to 0.5 for

TABLE 2—RESULTS OF TESTS OF STIFF FRAMES

	With Standard Frame			With Test Frame		
Car	Weight, Lb.	Rigidity, Lb. per In. Deflection	Front-End Shake, Average of Radiator and Lamps	Weight, Lb.	Rigidity, Lb. per In. Deflection	Front-End Shake, Average of Radiator and Lamps
A	267	460	0.43	214	860	0.19
B	212	305	1.10	207	1000	0.19
C	310	400	0.52	292	910	0.22

a 5½-mile run at 60 m.p.h. We had one car with a frame constant of 239, and it was not entirely satisfactory. Other cars having values around 180 were very satisfactory at first. I say "at first" because our ideas about what is a satisfactory job are constantly changing. In another year we may try to bring all of our cars to a point where the frame constant is not more than 175.

Our experiments have included a number of frames having a rigidity factor of 800 lb. or more, as determined by the standard rigidity test. In one case this factor was more than 1000 lb. Several of these stiff frames have been built into cars and road-tested, with results given in Table 2.

The figures show that the stiff frames produced a better condition of front-end shake, other factors being unchanged. Each car was first tested with a standard frame, then was torn down and reassembled on the test frame, using identical springs, bodies, shock-absorbers, engine mountings and equipment. The front-end shake was determined by wabblemeter test as described.

Value of Crankshaft Counterweights

MAXIMUM loads are created directly by a combination of inertia and dynamic forces and indirectly as their magnitude forces deflection of the sustaining parts, thus decreasing the bearing surfaces in contact and hence increasing the load. Counterbalancing the dynamic forces is the most important fundamental in the reduction of direct and indirect loading from this source.

The importance of counterweights in engine operation is not thoroughly appreciated. Only a few years ago they were used primarily as an accessory to smoothness, the general belief being that the weight must be proportioned to the reciprocating parts, that they served to cancel out the effect of these parts and that thus a degree of smoothness was brought about. Later, after a better understanding of the problems involved was acquired, counterweights were considered as no aid whatever to smoothness but merely as serving to take the load off the main bearings. They were taboo at that time because they increased the moment of inertia of a crankshaft and thus brought the torsional period well within the operating range of the engine. Today, the operating range has been increased to such an extent that harmonizers or torsional dampers are necessary, regardless of counterweights. Counterweights have therefore been rediscovered as an aid to bearing life.

We understand now that counterweights serve both to lower the load on the main bearing by their counteracting force and also to relieve the crankcase and the bearing supports of that portion of the dynamic load which is counteracted. This reduction in deflection of the crankcase and bearing supports does two very important things: (*a*) reducing the deflection keeps the journal and its supporting surfaces more nearly parallel, thus spreading the load over a greater surface, and (*b*) a definite improvement in smoothness is attained in proportion to reduction in the deflection of the crankcase, because this deflection is the major factor in high-speed roughness.

Item (*a*) is proved by the fact that the bearing improvement resulting from the use of counterweights is more than proportional to the reduction in load. Item (*b*) is indicated by the fact that operating harshness is reduced considerably when counterweights are used. Counterweights offer a high degree of result per dollar.

Counterweights are an effective aid to the main bearings of a crowded engine. The crankpin bearings, however, are not aided in the least. For reduction of the crankpin loads, we must use lighter reciprocating parts such as aluminum pistons and connecting-rods. Lighter reciprocating parts are an aid to the main bearings also, but they compare in no way with counterweights in result per dollar.—From Annual Meeting paper by Alex Taub, M.S.A.E., development engineer, Chevrolet Motor Co., Detroit.

Supplying Clean Oil for Engine Block Tests

THE service that should be expected from properly designed engine bearings is in many cases not obtained because of the lack of attention to the very important detail of keeping out of the oil all particles of foreign matter such as core sand, iron dust from cylinder-walls and piston-rings and other metal chips. Neither any amount of washing nor liberal use of air or gasoline sprays, oil-filters or other methods of cleaning can possibly keep destructive abrasives out of the oil during the initial run if the engine is run in with its normal lubrication system.

During the first hours of running under its own power, a surprising quantity of foreign matter is dislodged by vibrations in the whole engine structure, by increased temperature and by the wearing in of the bearings, pistons, rings and other moving parts. Some of the particles are so small and light that the ordinary oil-filter fitted to the engine is useless for removing them.

A number of engine builders have realized the importance of clean engines and have worked out methods that prevent the engines from being worn out before they start service. The method used at one plant is as follows:

When the engine is assembled, a pressure pipe from an outside oil supply takes the place of the regular connection to the oil manifold from the oil-pump in the engine. This connection passes through the drain hole in the oil-pan, the drain plug being removed. The outside oil system consists of large centrifugal separators and settling tanks fitted with suction pipes to draw oil from the oil-pans and pumps to supply oil under pressure from the tops of the settling tanks to the test stands.

Test stands are arranged in pairs, so that each green engine is driven for 4 hr. by another engine that has already had its initial 4-hr. run under power from a previous engine. The bearings are inspected after the first 4-hr. run before the engine is run under its own power. After the second run of 4 hr., the regular oil line from the pump to the manifold is connected and the engine is given a short run to make certain that the pump is functioning properly.

Under this system the engine is run 8 hr. under varying conditions of load and temperature with oil that is even cleaner than new commercial oil. Such a system contributes far more to the ultimate life of the engine than does supplying an ordinary oil-filter that requires a refill every 10,000 miles, as virtually all damage from foreign particles is caused during the first few thousand miles of service by the abrasives that are built into the engine, while service records show that only a small percentage of the oil-filters are ever renewed by the car owners.—From Annual Meeting paper by D. E. Anderson, M.S.A.E., of the Bohn Aluminum & Brass Corp., Detroit.

The Use of Trailers with Motor-Trucks

Semi-Annual Meeting Paper

By James W. Cottrell[1]

TRAILER registration figures for the entire United States are given to show the rapid increase in the use of trailers in the last seven years, and, for comparison, State registrations of all motor-vehicles in 1931 are given.

To account for the relatively more rapid increase in trailers than in trucks, factors favoring the use of trailers are mentioned and illustrative examples of operation are briefly described. The factor of first importance is legislation, which in general is stated to have promoted the use of trailers to distribute the weight of heavy loads over more axles and wheels; but in some States the laws and regulations have a serious adverse effect.

Next to legislation, savings in hauling costs through the use of trailers account for the increase in their numbers, and comparative figures of the cost of hauling per 100 lb. per 100 miles by truck, by truck and trailer and by rail are given to show the economy.

Another important factor mentioned is convenience to shipper and consignee, and examples of the services rendered are cited. Operators are stated to foresee the development of a Nation-wide service with interchangeable trailers much as railroad freight-cars travel to all parts of the Country over various railroads. Before this can be realized, coupling mechanisms must be standardized so that any trailer can be coupled to any tractor. Difficulties that stand in the way of accomplishing this by a subcommittee of the Society are set forth.

Trailer designers have been confronted with a difficult task in designing the semi-trailer fifth-wheel coupling, which must satisfactorily meet a multitude of requirements, but they have succeeded in refining and reducing the weight of this mechanism and also of the trailer body.

SURPRISING, is it not, to be reminded at this late date that we lack a clear-cut definition of the term "trailer"? Yet no less an authority than Thomas H. MacDonald, chief of the Bureau of Public Roads, reminded the Eastern Conference of Motor Vehicle Administrators in the City of Washington on May 5 and 6, 1932, of the omission. He said, "We should take a long step forward if clear understanding could be reached of exactly what we mean when we say 'truck' or 'bus'; 'tractor', 'trailer', 'semi-trailer', 'single unit' or 'combination of vehicles'." And, speaking of overall length limits, he added:

> An adjustment of an important definition may help us to a decision regarding this question. If a tractor with a semi-trailer is to be considered as a single unit and trailers are to be permitted, 65 ft. is undoubtedly the necessary length for an economical combination of vehicles. If, however, a tractor with semi-trailer is itself to be considered a combination of vehicles . . .

This confusion does not arise in England, where a tractor and semi-trailer are regarded as one vehicle and designated "an articulated six-wheeler" to distinguish it from a "rigid six-wheeler"; that is, our six-wheel truck.

Leaving to others the problem of nomenclature and definition, in this paper, when referring to trailers, I shall include semi-trailers and full trailers, the former being vehicles towed by and in part carried by tractor power units and the latter denoting independent vehicles with four or more wheels towed by but not resting upon the power unit. How to classify semi-trailers with front dollys or a proposed design of semi-trailer with a booster engine driving the semi-trailer axle only on steep grades, like the booster engines carried under heavy-duty locomotive tenders, may remain, for the time being, an unsettled question.

Rapid Growth of Trailer Registrations

Despite the oversight of lexicographers, trailers are being used in quantities and their use with trucks is growing rapidly. Trailer registrations at the close of 1931 showed an increase of 27 per cent over 1930 and 300 per cent over 1927. Figures for total truck registrations show no parallel; a slight decrease occurred between 1930 and 1931, and the gain between 1927 and 1931 was only 14.9 per cent.

Total registrations of trailers at the close of each of the last seven years were:

1925.....	83,625	1928.....	151,596
1926.....	99,430	1929.....	197,290
1927.....	123,451	1930.....	261,913
		1931.....	352,495

Some reservations about the accuracy of these figures are in order. First, not all of the States register trailers separately. More States registered trailers, as such, in 1931 than ever before, but even in that year four States and the District of Columbia gave no figures for trailer registrations. Estimates for the missing States indicate the total number of trailers in use as 370,000 rather than 352,495.

The second variable is the fact that the States registering trailers do not follow the same definition of a

TABLE 1—TRUCK AND TRAILER REGISTRATIONS IN 1931

State	Trucks	Trailers	State	Trucks	Trailers
Alabama	33,895	3,279	Nebraska	59,848	15,737
Arizona	12,633	1,623	Nevada	6,950	411
Arkansas	22,000	2,800	New Hampshire	18,671	1,137
California	105,213	59,057	New Jersey	135,098	2,916
Colorado	32,082	258	New Mexico	15,677	802
Connecticut	53,274	1,062	New York	347,443	13,250
Delaware	9,991	517	North Carolina	54,425	8,268
District of Columbia	19,809	....	North Dakota	26,588	....
Florida	50,819	6,751	Ohio	191,929	32,717
Georgia	45,736	3,317	Oklahoma	54,585	....
Idaho	15,435	7,558	Oregon	24,288	2,361
Illinois	201,509	9,283	Pennsylvania	219,696	6,308
Indiana	129,192	17,807	Rhode Island	20,112	90
Iowa	80,752	3,207	South Carolina	24,724	2,100
Kansas	80,484	1,776	South Dakota	23,816	4,988
Kentucky	36,670[a]	....	Tennessee	35,096	2,696
Louisiana	47,783	5,445	Texas	210,850	33,798
Maine	35,711	3,084	Utah	17,577	767
Maryland	35,904	1,126	Vermont	8,453	457
Massachusetts	103,888	650	Virginia	58,991	1,582
Michigan	152,635	61,932	Washington	61,114	3,000
Minnesota	109,984	18,329	West Virginia	38,907	1,270
Mississippi	30,721	2,560	Wisconsin	118,223	1,007
Missouri	96,000	5,356	Wyoming	10,917[a]	....
Montana	24,037	52	TOTALS	3,350,135	352,495

[a] Trailers included with trucks.

[1] M.S.A.E.—Technical editor, *Commercial Car Journal*, Philadelphia.

trailer. The light two-wheel camp-type vehicle pulled by passenger-cars may or may not be a trailer, according to law; some States require a trailer license for an axle and pair of wheels on which a load of telephone poles is balanced, and so it goes. Trailer registrations for the respective States therefore include whatever vehicles the State authorities register as trailers.

Some States Are Trailer-Minded

The increasing use of trailers without a corresponding advance by trucks signifies greater acceptance of the trailer idea and that the ratio between the number of trucks and of trailers is becoming more favorable to the latter. The ratio between trucks and trailers varies in different States, as a study of the figures in Table 1 shows.

Table 2 shows the five States that lead in trailer registrations. California, second in total number, also leads in percentage of trailers, having 0.56 trailers per truck. New York State, with the largest truck registration, which is more than twice as many as in Michigan, has only 21 per cent as many trailers.

Why does California use proportionately 10 times as many trailers as New York State, or 5 times as many as the National average? The answer is to be found in the answer to the general question, Why use trailers? In other words, what factors determine whether trailers can be used to advantage? Legislation ranks first, because, if trailers are prohibited, no other factors enter. Then are to be considered grades and road conditions, characteristics of loads to be hauled, length of haul, loading and unloading time and facilities, time available from shipping platform to receiving platform and convenience of the shipper.

Psychology, too, has exerted an influence; some might call it a spell. The state of mind of those who sell and those who use highway hauling equipment obviously influences trailer application. Not so many years ago many argued for or against trailers, as an abstract proposition; others simply disregarded them altogether except when loads to be carried were beyond the capabilities of any truck that could be operated under the laws and regulations existing at the time and place. A transportation engineer of wide experience recognizes the importance of this attitude of transportation men toward trailers when he states:

> We might cite Michigan, Ohio and California as States which are trailer-minded and are using trailers in almost every industry and for every occupation.

Legislation Promoted Trailer Use

Legislation and regulation helped the trailer to get its start and win its place in highway transportation; not that the lawmakers were inspired particularly by kindly regard for the trailer but rather that they sought to protect the roads. But as soon as they allowed greater loads on six or more wheels than on four, just so soon did they invite the use of trailers. Fred B. Lautzenhiser, of the International Harvester Co., states:

> More and more each year, State legislation is making trailer use imperative by the restrictions placed upon total weights on individual axles. At the same time, some States are placing maximum dimensions on over-all train lengths, which, in conjunction with the limited axle weights, makes interstate hauling a difficult problem.

Difficult indeed, because trailer regulations not only are not uniform but are constantly changing. The compilation of State laws and regulations made last year by the Motor-Vehicle Conference Committee showed that 19 States had no direct restrictions on the number of trailers that could be used in combination, and, of the 30 States having such restrictions, 19 permitted only one trailer, 5 permitted a tractor and a semi-trailer to tow one other vehicle, and 6 permitted two trailers behind a motor-vehicle unit. This indicates that a four-wheel trailer making a long interstate trip may be stranded more than once.

The most recent example of a change in legislation is the action of the Kentucky legislature, which, by a new act barred all full trailers after June 18, 1932, and, by another provision, effective Jan. 1, 1933, reduced the weight limit of a tractor and semi-trailer to 18,000 lb. and the length to 30 ft.

TABLE 2—MOTOR-VEHICLE REGISTRATIONS IN 1931 IN STATES REGISTERING THE LARGEST NUMBER OF TRAILERS AND IN THE ENTIRE UNITED STATES

	Passenger-Cars	Trucks	Trailers	Trailers per Truck
Michigan	1,078,345	152,635	61,932	0.40
California	1,971,170[a]	105,213	59,057	0.56
Texas	1,084,624	210,850	33,798	0.16
Ohio	1,531,000	191,929	32,717	0.16
New York State	1,928,701	347,443	13,250	0.038
United States	22,550,077	3,350,135	352,495	0.105

[a] Includes approximately 134,000 commercial cars under 3,000 lb.

Such differences and changes introduce an element of uncertainty into both interstate and intrastate hauling. For illustration, a contract hauler who bases his costs on the use of trailers may be in a serious position if legislative restrictions stop him from using trailers or reduce the weight of the load he may carry on a combination of vehicles. Private owners are not immune. Consider the loss to an oil company if its distribution set-up, based upon existing truck legislation, is rendered inoperative by later enactments; capital investment in additional distributing stations may reach large figures.

What other factors decide the case for or against trailers?

Boundaries for the field of use of trailers may be located, although they are not clear-cut. For certain types of work the need for trailers is obvious; for others, the opposite is true. In many cases either trucks or trucks and trailers can be economically employed. Interviews with fleet owners, factory engineers, State highway officials, dealers and others show that they are studying the question and accumulating information upon which to base decisions.

Roads and grades are a limiting factor in trailer application, in the opinion of an engineer on the staff of a factory making both trucks and trailers. He states that

> trailers require improved roads and fairly level country. . . . To triple the rated pay-load of a truck and expect it to develop satisfactory performance over an unimproved highway is out of the question.

Under the conditions he specifies, he allots the trailer a good share of all the work, saying:

> For most city work and where good roads are to be found, the trailer can be used for about 90 per cent of all of the general hauling.

Hauling Costs Reduced with Trailers

Lower cost of hauling doubtless is the reason for the use of trailers in most applications and is an argument that needs no elaboration. Given conditions suitable for trailer operation, a reduction in costs may be expected for the same reason that three-horse plows, 100-car freight-trains, double-deck omnibuses, huge ocean liners and 16-passenger transport planes are used to increase the volume of work accomplished per man.

Substantial savings resulting from the use of trailers

with trucks were recently pointed out to the Transportation Club of Toronto by F. J. Scarr, who also demonstrated the effect of these savings upon the competitive position of motor hauling compared with rail freight. In a chart of comparative freight transportation costs in l.c.l. door-to-door service he gave the cost of hauling 100 miles by 5-ton truck as 46 cents per 100 lb.; by 5-ton truck and trailer, 33 cents; and by a combination of tractor, semi-trailer and full trailer, 25 cents, the last figure being a reduction of practically 46 per cent from the truck cost.

The truck figure of 46 cents is more than that for container-car service for 100 miles, the truck and trailer figure is less than container cost at 100 miles and equals it at 130 miles, but the cost by tractor, semi-trailer and full-trailer train is less than the container-car cost at all distances from 10 to 200 miles.

In published estimates by a trailer engineer, similar costs are given as 45 cents per 100 lb. for truck hauling, 29 cents for tractor and semi-trailer and 19 cents for tractor, semi-trailer and full trailer.

With these savings as incentives, it is small wonder that the use of trailers increases. In addition to direct savings in dollars, there are various other advantages that cannot be figured so readily in dollars and cents. A few examples of various trailer applications will show savings in both time and money. These may be classified roughly into city service, intercity service and long-distance operations.

Store-Door and Rail Service Rendered

The Chicago office of the Railway Express Agency retired its last horse-drawn unit in April, 1932, thus completing motorization of this office. When the express companies were consolidated into the American Railway Express Co. on July 1, 1918, about 1800 horses were in use in the Chicago district. The present fleet consists of 542 trucks, 65 tractors and 166 trailers. Approximately one-third of the tractor-trailer fleet is operated on double shifts and about one-sixth on triple shifts. The ratio of just about 2½ trailers to 1 tractor shows the importance of loading and unloading in this operation.

Another example of trailer operation in connection with rail service is the "ferry truck" installation in Chicago. The ferry trucks are trailers owned or operated by the Chicago, North Shore & Milwaukee Railroad. Shippers desiring to use the service notify the railroad and the carrier places the ferry trucks at establishments in the cities of Chicago, Racine or Milwaukee, where they are loaded with l.c.l. freight to be forwarded to consignees at points served by the railroad or its connecting lines. Shippers who do not desire to use the ferry trucks may deliver freight at the freight stations in their own vehicles.

The ferry trucks or trailers are hauled by tractors to the rail stations, where they are loaded upon specially constructed flat-cars, some of which hold two trailers, while the new type carries three trailers. The flat-cars are equipped with devices to lock the trailers in position. At destination the ferry trucks are unloaded from the flat-cars, attached to tractors and delivered to consignees' platforms. According to G. Lloyd Wilson, professor of commerce and transportation at the University of Pennsylvania:

> The ferry truck obviates the necessity of rehandling the freight from the time it is loaded into the trucks until it is unloaded at the consignee's door. Savings are possible also in the elimination of markings upon packages and, in some cases, in the amount of packing required to protect the goods.

Truck, trailer and railroad interests are known to be considering the establishment of similar service elsewhere. No revolution in transportation is required to render the service. Railroads possess an abundance of flat-cars, and locking devices for trailers need not be costly or complicated. Trailers and tractors need not be owned by the railroads. They can perform the pick-up and delivery and the railroads can perform the line haul.

A third example of trailers handling rail shipments is the transfer of l.c.l. freight between freight stations of different railroads at a rail center. The Texas Warehouse Co., of Fort Worth, put a tractor and six semi-trailers into this service several years ago and, according to Nielsen survey figures, this shuttle service saves 42 cents per ton, or 37.8 per cent, compared with straight truck operation.

Store and Dairy Distribution Work

Many large department stores use semi-trailer shuttle service to carry packages in bulk from downtown stores to remote delivery stations. A typical example is the service of a large department store in Newark, N. J., which loads semi-trailers in a sub-basement and hauls them with tractors to a remote delivery station. An important advantage of the tractor-semi-trailer combination is its ability to maneuver around posts and make a backing turn from the freight elevator to the loading platform.

Insulated bodies on semi-trailers are used by a large dairy in the Middle West as branch distributing stations. Two such stations are maintained, each serving regular delivery routes. Two semi-trailers and one tractor are used, and an ingenious working out of the movement of the three units obviates dead mileage. The tractor starts in the morning from Station *A* with a semi-trailer, which I shall designate *A*, carrying cases of empty bottles to the downtown bottling plant, where the cases are removed and filled cases loaded. The tractor then takes Semi-Trailer *A* to Station *B* and places it beside the loading platform. Here Semi-Trailer *B*, also loaded with cases of empty bottles, is picked up and returned to the bottling plant, where loads are exchanged as before and the tractor with loaded Semi-Trailer *B* proceeds to Station *A* for a night lay-over.

The cost of interplant hauling and of pick-up and delivery service to freight stations was reduced about $35,000 per year by a large factory through a change from all-truck transportation to a few trucks and a number of trailers that do 90 per cent of the hauling. This figure is based on operation in a time of prosperity, as in 1929, and no doubt the volume is considerably less at this time.

Intercity and Local Work Combined

Demands of automobile and truck factories for deliveries of parts and units on schedules keyed to assembly-line requirements have inspired the use of trailers for hauling between suppliers' plants and the motor-vehicle factory. J. R. Caldwell, who has been hauling tires between Akron, Ohio, and Detroit for more than 10 years, uses a powerful heavy-duty tractor pulling a two-wheel semi-trailer and a four-wheel trailer. The combination is operated like a through railroad train, both semi-trailer and trailer being loaded and unloaded by terminal crews and smaller tractors transporting the loads in trailers to destination after the train arrives in Detroit.

Local pick-up and delivery service frequently is tied in with intercity trailer operation. If a trailer carries a 10-ton load made up of a number of smaller individual shipments it may be profitable to pick up these shipments in trucks and load them in the trailer at a central

loading platform rather than to drive the tractor and trailer about town picking up the load. Obviously, the same applies to delivery.

Speed Becoming an Important Factor

Trailers are now operating in intercity service at speeds that have caused many operators to think about wind resistance. Round-front-end bodies were designed primarily to increase the amount of loading space available ahead of the king-pin but the possibilities of reducing wind resistance have not been overlooked. A semi-trailer body pointed at both front and rear was recently put in service between Newark, N. J., and Philadelphia. The unit is used to relay an average load of 9 tons of pies from the Newark plant to the Philadelphia distributing station. The body is 20 ft. long, 7 ft. wide and 6 ft. high, has sides and top of aluminum and is 40 per cent lighter than a conventional body. Stanleigh Megargee, supervisor of automotive equipment of Pie Bakeries, Inc., who designed the body, states that it greatly reduces wind resistance without sacrifice of cargo space. He is making wind-tunnel tests of the body and expects to get some very interesting figures.

The basic trailer idea used so extensively for general hauling is equally applicable for specialized hauling as of milk, oil and heavy machinery.

With the cooperation of tank manufacturers, trailer builders have evolved both semi-trailers and full trailers for hauling a wide variety of liquids. In these operations, loading and unloading are practically automatic and designers concern themselves with the problem of carrying a given quantity in the most economical way. Oil refiners use drop-frame semi-trailers to carry tanks for hauling to retail stations and, by increasing the load of such a unit, have extended the length of the routes served. For bulk hauling of gasoline, trailers have been built which carry unit loads up to limits set by State legislation. Weight has been saved and legal load capacity increased by the use of aluminum for tanks and parts of the trailer chassis and in some cases by using the tank as the main structural unit, thus eliminating a separate frame. A single semi-trailer carrying 5000 gal. of gasoline was, when it appeared, looked upon as an engineering achievement. A recent typical design of aluminum tank on a semi-trailer pulling a four-wheel trailer carries just a few gallons less than 6000 gal., and this capacity will be exceeded when legislation that will not be changed overnight permits increased total gross weight of vehicle and load.

Legislation based upon axle-load limits is favorable to trailer operation provided the total length permitted is not too short. A train operated by the Detroit Creamery, which illustrates this point, has an axle-load limit of 18,000 lb., and a total gross weight of 99,000 lb. is moved as a unit. The train is comprised of a four-wheel truck and two four-wheel trailers on which 1250-gal. stainless-steel tanks are installed.

Large-Capacity Van Bodies Used

The bulk of furniture in relation to its weight requires relatively large space for a truck load. When van capacity reached 1000 cu. ft., the question of load space on the truck became important. A semi-trailer chassis provides the longer effective wheelbase and greater load space required without resort to an excessively long truck frame. Vans of 1200 or 1400-cu-ft. capacity are able to carry the entire furnishings of two six-room apartments, including pianos, in a single load.

Vans are not all of large capacity; much moving is being done by semi-trailers drawn by tractors of the 1½-ton low-price variety. In the smaller cities, where there may not be enough moving to keep a van busy every day, household furniture is being moved successfully in semi-trailers fitted with rack body and covered with tarpaulins. One commercial hauler who tried this reports that furniture arrives in better condition under tarpaulin than in regular vans and that the saving in weight of body is considerable.

Van-type bodies are used for specialized service as well as for general freight. Carriers for show horses and race horses are becoming more and more popular. In fact, a horse show has enough motor equipment back stage to interest any truck man.

Semi-trailers with drop frames provide a lower floor than do standard trucks, which is an advantage in certain operations, such as tank and van work.

Operators Visualize Nation-Wide Service

Operators of vans have suggested, through their cooperative organization, the extension of hauling by trailers to Nation-wide proportions, shipments being made between any two places in the Country. Some foresee trailers moving about from place to place like railroad freight-cars. The idea is attractive but some major problems must be worked out before it can be put into effect. Conservative transportation engineers content themselves for the present with speculating about a gradual extension of trailer operation from existing trailer routes.

When time is the essential element in shipping, trailers can be used for relatively long hauls. Florida strawberries were hauled to the New York City market last spring by tractor-semi-trailer units, making the trip in 48 hr., which is approximately one-half of the total time required for rail shipment. According to the operators of this line, they secured the business at a price which gave them satisfactory profit. The tractors carried sleeper cabs in which the drivers alternated on 6-hr. shifts. No stops were made on the way except for fuel and oil.

Trailers provide the only practical means for hauling freight that is too bulky or too long for trucks, such as structural-steel assemblies, scenery, pipe and poles. A large load of boxes or similar freight may require too much space if loaded on a truck. Even if it is possible to get the load on an extra-long-wheelbase truck, the difficulty of maneuvering in traffic or in close quarters near loading platforms remains.

Other factors also must be considered in deciding whether to use trailers. A truck sales engineer of long experience considers very carefully in all of his analyses the length of haul and the time available for a trip. When working out a hauling problem, he asks the shipper about the demand for quickness in delivery and also the length of the working day for drivers and helpers. Explaining the importance of these factors, he said:

> The most expensive item in hauling that is under our control is personnel. We work out transportation equipment to use all of a man's time. If a truck can make a round trip in 8 hr. out of a 10-hr. day, the extra 2 hr. may be lost. In a day of 10 hr. we might make the trip with a truck and a four-wheel trailer, thereby cutting hauling costs because the labor charge is no more and the load hauled is much greater.

Support and Movement of Load Involved

Transportation by motor-vehicle involves two separate functions: supporting a load and moving it. The argument, frequently used on behalf of trailers, that a truck can pull more than it can carry, does not apply in all cases, because of the restrictions imposed by road

conditions, grades and legislation. An engineer, speaking of this argument, advises that the statement be changed. He puts it, "a truck may, under given operating conditions, be able to pull more than its frame structurally can support." The semi-trailer provides additional support at its rear axle.

The function of a full trailer is to provide additional load capacity, and it may be used to add capacity to a four-wheel truck or to a tractor and semi-trailer unit. The question frequently arises whether a truck and full trailer or a tractor and semi-trailer should be used. A factory engineer, asked for his opinion, stated that the answer depends upon local conditions. He explained that statement as follows:

> We recommend that an analysis be made. Legislation enters into the question; for example, in Illinois, even though a tandem-wheel semi-trailer and a four-wheel tractor would be the ideal unit, the law is so written that a four-wheel truck and a four-wheel trailer permit the greatest pay-load. If the equipment must deliver or pick up in congested places, the four-wheel trailer should not be recommended, as it is hard to back up. Therefore, for cities and local hauling, the semi-trailer makes the ideal recommendation. For long-distance work, where the time lost in spotting a four-wheel trailer may be only a small fraction of the total running time, it may be more economical to use the four-wheel trailer provided it permits more pay-load; but as a rule the semi-trailer represents the ideal trailer installation.

Advantages of Trailers for Small Loads

Although most of the preceding discussion has been devoted to the use of trailers to carry greater loads than can trucks, trailers are not confined to this field. On the contrary, one of the striking developments of the last few years has been the use of semi-trailers to carry loads well within the capabilities of four-wheel trucks. With low-priced tractors available from the large-production 1½-ton trucks, the semi-trailer advocates present a strong plea for carrying loads of 3 to 5 tons. The first advantage they point out is that of reduced first cost, although reduction in truck prices in the medium-duty class makes this margin less than was the case a few years ago. A. W. Kennerson, of the Standard Oil Co. of Ohio, put the saving in first cost at approximately $1,000 in February, 1930, when he said:

> When you take into consideration the fact that we find it possible to increase the capacity of our units 100 gal. at an initial saving of from $800 to $1,000 per unit compared with the average 2-ton truck, the answer is obvious.

Use of semi-trailers for hauling loads of 3 to 5 tons has had the important effect of diverting attention from the trailer exclusively as a means of carrying greater maximum loads and directing it to a greater extent to the proper sphere of the trailer in highway transportation. There is ample evidence that all parties concerned have benefited and will continue to benefit by this change.

Convenience to shippers is a factor that is not to be overlooked in the use of either light or heavy trailers. Trailer bodies can be spotted along a factory delivery platform in any number and removed on a few minutes' notice, even though actual shipping time may be several hours later. The same set-up is employed in motor freight terminals, where freight coming in from various pick-up routes is sorted directly into trailer bodies, which serve as sorting bins and thereby save rehandling. This advantage to shippers aided the development of heavy-duty trailer applications and is also helpful in the lighter-duty trailer field; in fact, it may decide the question in favor of semi-trailers for loads that can easily be hauled by a medium-duty truck.

Advances Made in Design

Contributing to increased use of trailers with trucks is the advance in design and construction of trailer chassis and bodies. In the early period of trailer use, less attention was given to engineering in the trailer itself than at present. Operators who were anxious to take advantage of the load-carrying ability of trailers or their economy or both, devoted less attention to weight, cost and operating expenses than they do at present. Successful operators who maintain accurate cost systems know exactly how much it costs them to operate trailers. They know the value of weight saving in load-carrying vehicles, they look for some interchangeability between truck and trailer, and they expect much of trailer manufacturers just as they expect much of truck makers.

Trailer designers have been able to make trailer chassis lighter and stronger. They are prepared to furnish trailers with tires and wheels matching the tractor, and in some cases they supply other units that are interchangeable with tractor units. Cooperation between trailer maker and body builder has resulted in simplified design and reduced weight, cost and operating expense. Trailer brakes are one element showing engineering advance in design and the effect of legislation, which is setting up strict requirements for trailer brakes in several States. Some States require the same stopping ability for a tractor and trailer unit as for trucks; others set up severe limits expressed in distance required to stop from a specified speed. A number of States specify how trailer brakes shall be operated; for example, Florida requires trucks and trailers operating under certificates to be equipped with "driver control" air or vacuum booster brakes, and Wisconsin calls for brakes operated by power from the driver's seat or "by an adult attendant on such trailer or semi-trailer." Adult attendants are something new in this Country, although "volunteer" riders on trailers are numerous.

Delaware allows trailers or semi-trailers having a gross weight in excess of 4000 lb. to operate without brakes provided the speed does not exceed 10 m.p.h., which is just a roundabout way of forcing trailer owners to use brakes. According to the Motor Vehicle Conference Committee, "adequate" brakes on trailers are required in Colorado, Connecticut, Illinois, Kansas, Maryland, New York, North Carolina, Utah and Wisconsin, while California and Oregon require ability to stop in 37 ft. from 20 m.p.h. Other States specifying stopping distances from 20 m.p.h. are: Pennsylvania, 50 ft. with both hand and foot brakes or 75 ft. with either; Virginia, 30 ft. for both sets of brakes and 50 ft. with either set; and Texas, 45 ft.

Difficult Functions of Fifth Wheel

The mechanism that couples a semi-trailer to a tractor, commonly called the fifth wheel, has been and still is one of the most important factors in tractor development. Even in its simplest form, that between a tractor and semi-trailer which are not uncoupled in operation, it is subjected to stresses, impacts and contortions of a most complex nature. Steering calls for provision for angularity between tractor and trailer frames in a horizontal plane; irregularities in the road and grades call for angularity between the frames in a vertical plane; if a rear wheel of the tractor drops into a hole, an angle between the frames is created in a third dimension, and all three angles may exist at the same time. Add to this the duties of supporting the weight of the front of the semi-trailer, pulling the entire

weight along the road and, perhaps, holding it back on hills, and then demand silence, long life, a cushioning effect and light weight—have the designers not been set a task?

But this is not all. If the semi-trailer is to be coupled and uncoupled, an automatic latch must be provided to enable the tractor to be coupled to the semi-trailer while it is supported on its dolly wheels, which are raised and lowered by hand. This latch must be simple in design, rugged in construction and reliable in operation. High speeds and heavy loads are destructive of king-pins and locks, and even a slight degree of looseness or play sets up a hammering action like the pounding of a loose connecting-rod bearing. Therefore the latch or lock must be tight without binding.

Still another job for the coupler is that of automatically lowering the dolly wheels to support the semi-trailer just before uncoupling and of raising them when coupling. This added function distinguishes the fully automatic coupler. To uncouple the tractor from its accompanying semi-trailer, the driver pulls a lever and slowly drives the tractor away. To couple, he backs the tractor under the semi-trailer and pulls away. The ease with which the coupling and uncoupling are accomplished hides the fact that the coupler performs a number of functions in sequence.

Non-Interchangeable Couplers a Handicap

Because the coupling is easy and its usefulness manifest, a strong and growing demand has arisen that all couplers be interchangeable, as is the case with railroad-car couplings. Trailer operation is hampered by the fact that fully automatic couplers and even plain couplers are not interchangeable. The need for interchangeability is obvious and has been discussed time after time. A short time ago Robert H. Dunn, chairman of the Michigan Public Utilities Commission, emphasized the disadvantage of lack of interchangeability at a meeting of the Central Motor Freight Association in Chicago.

A demand of fleet operators for action looking toward standardization led to the appointment of a subcommittee of the Society to study the subject, and the subcommittee invited operators and trailer manufacturers to confer. At the conference a general agreement was reached that the king-pin and lock of couplers might be standardized but the fully automatic couplers were a stumbling block. Trailer manufacturers having established designs of fully automatic vehicles are loath to give up their main sales and talking points, which are the features of design of the coupling mechanisms. Among other things, patents stand in the way of standardization.

A truck engineer privately suggested that semi-trailer dolly or support wheels might be raised and lowered by power from the tractor, using compressed air, vacuum or electricity, or that the front end of the semi-trailer might be raised and lowered by a power arm or jack carried on the tractor. This would make coupling independent of forward motion of the tractor and would also make both coupling and uncoupling possible regardless of any angle between tractor and semi-trailer.

A solution of the standardized coupler has not yet been found, but the matter is not closed.

Makers have reduced the weight of fifth wheels by carefully designing each part and generally refining the design. At least two trailer manufacturers have introduced rubber insulation to reduce shock, noise and vibration.

In conclusion, I desire to express my appreciation for the cooperation by trailer owners, manufacturers, dealers and State officials in the preparation of this paper.

While the use of trailers might be discussed at much greater length with interest and advantage, perhaps this cursory paper will suffice to show that the trailer is capable of carrying a large share of highway freight and to accept the general responsibility of a partner of the truck.

THE DISCUSSION

R. B. JONES[2]:—By far the most important factor affecting the use of trailers is the legislative action against them in many States. This works great hardship on the operators and manufacturing companies and has been particularly noted by manufacturers where equipment is sold on time-payment or finance plans. The fact that the laws are constantly changing in many States often requires the operator to move the equipment from one State to another or to rebuild it to comply with regulations. In some cases the operators turn the equipment back to the manufacturers because of this.

Until a more uniform code has been developed and is adopted by the majority of States, the savings and advantages of trailers to the public and to operators, including the railroads, will not be realized.

The reason that the subcommittee of the Society has not accomplished anything definite in the standardizing of interchangeable coupling mechanisms is that the manufacturers have not had any uniform understanding as to the requirements to be met in the functions that the fifth wheel should perform. Speaking in behalf of our company, I feel that much good would come from a universal fifth wheel and believe that the trailer manufacturers would welcome a standardized fifth wheel provided they knew what part they would play and how they were going to be treated if a standard were adopted.

The problem, instead of being simplified at present, is becoming more complicated because other manufacturers are attempting to make still different types of fifth wheel than those already existing, adding to the number of parties that have to be reconciled in the adoption of a standard. I believe that ultimately we shall have a standard fifth wheel and that it will be very much to the advantage of the trailer companies. I feel that the Society should continue the work it has started on fifth-wheel standardization, and that it will be greatly to its credit to complete it.

JOHN WALKER[3]:—While the Society may be primarily concerned with engineering developments and requirements and the coordination of these efforts, the present situation demands a broader view by the industry as a whole.

Mr. Cottrell quotes registration statistics that show a marked increase in the number of semi-trailers in use each year. This fact is emphasized when we compare truck registrations for 1931 with those for 1930, there being no increase of truck registrations as compared with approximately a 50-per-cent increase for trailers. This condition is not the result of high-pres-

(*Concluded on p.* 344)

[2] M.S.A.E.—Chief engineer, Trailer Co. of America, Cincinnati.
[3] Engineer, special equipment, Mack Trucks, Inc., New York City.

Piston-Ring Progress

Semi-Annual Meeting Paper

By Ralph R. Teetor[1] and Harry M. Bramberry[2]

THREE newly developed instruments—for piston-ring measurement, for ascertaining the radial pressures exerted and for determining "How rough is smooth?"—are described and their uses discussed. The last is a device to determine the smoothness of cylinder finish and predetermine piston-ring life. By all these means, the minute dimensions and characteristics of piston-rings can be studied and evaluated.

The authors call attention to the progress made in the last few years in piston-ring design, together with the important factors contributing to successful performance and long life of piston-rings. They state that considerable work has yet to be done before complete specifications for all diameters and types of ring can be given to the industry; however, the progress they describe has been made in ring equipment for high-speed engines under 4 in. in cylinder diameter. They believe that the fundamentals apply to piston-rings for all uses. Future developments will, in their opinion, make it necessary to deviate somewhat from present practice and make obsolete even the things that seem satisfactory today.

PISTON-RING equipment required for modern high-speed engines and testing apparatus for determining the major characteristics affecting the performance and life of piston-rings constitute the subject matter of this paper.

As engine speeds and compression ratios have increased steadily in the last several years, piston-ring equipment to meet such demands has required very different characteristics from those for the rings used in the lower-speed low-compression engine. It has necessitated an almost endless amount of effort and research, together with the development of gage apparatus by which the minute dimensions and characteristics of piston-rings could be determined. This work has been in progress for several years, particularly in the last two years.

[1] M.S.A.E.—First vice-president, in charge of engineering, Perfect Circle Co., Hagerstown, Ind.

[2] M.S.A.E.—Engineer, Perfect Circle Co., Hagerstown, Ind.

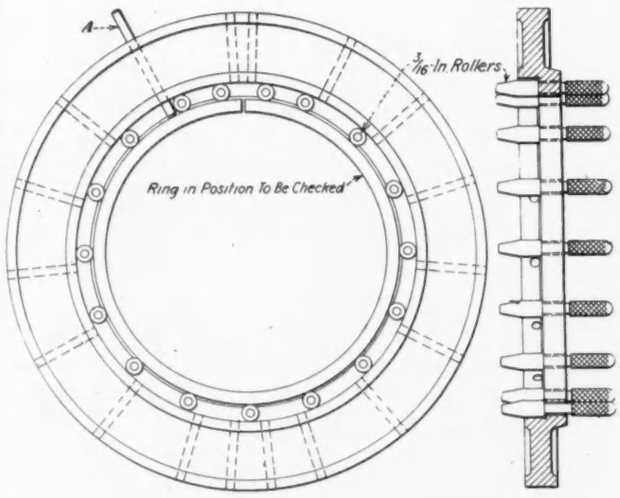


FIG. 1—ROLLER GAGE FOR TESTING PISTON-RINGS SHOWING IN CROSS-SECTION AT RIGHT THE TYPE OF ROLLERS USED

For Determining the Pressure Exerted by the Piston-Ring on Each Roller, 17 Plungers Similar to That Shown at *A* Are Located near Each of the 17 Rollers. These Are Inserted through the Roller Gage in the Manner Shown and Make Contact with the Piston-Ring as Indicated. The Knurled Shank of Each Roller Projects through the Roller Retainer So That It Can be Clasped by One's Fingers

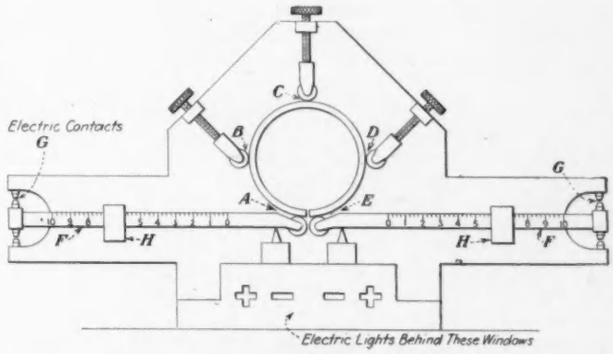


FIG. 2—ILLUSTRATION OF THE FUNDAMENTAL PRINCIPLES OF THE RADIAL-PRESSURE SCALE

A Roller Gage for Piston-Rings Devised

The "roller gage" shown in Fig. 1 comprises a solid steel ring having an accurately finished internal cylindrical surface, the diameter of which is greater than that of the ring to be tested by twice the diameter of the individual inserted rollers. The 17 rollers are located loosely in a retainer that is a portion of, or attached to, the steel ring; and are located so that they form an insert between the interior surface of the gage and the exterior face of the piston-ring. The rollers are spaced on 22½-deg. centers, excepting the two rollers that are located on either side and immediately adjacent to the joint. When the ring is inserted in the gage, a very practical measurement of the radial pressure of the ring against the rollers can be made by turning each roller between the thumb and finger. This gage provides the most practical method generally known at present, and has contributed considerably to the development of determining correct radial wall pressure.

A Radial-Pressure Scale Developed

The foregoing process is slow for checking radial-pressure characteristics; therefore, a "radial pressure scale" has been developed by which the pressure at the points of the ring can be measured quickly and accurately. Fig. 2 outlines the fundamental principles of this scale. There are five rollers, indicated by *A*, *B*, *C*, *D* and *E*, which are set in position by using a disc of a size equivalent to the cylinder diameter. Rollers *A* and *E*, mounted on the ends of scale beams *F*, contact with the ring just as close to the joint as possible while permitting a slight clearance between them. It is very

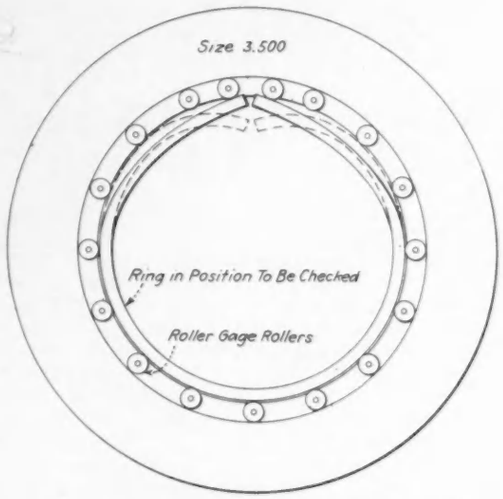


FIG. 3—DIAGRAM SHOWING THE ROLLER GAGE WITH TWO EXTREMES IN PISTON-RING SHAPE SHOWN IN EXAGGERATED FORM

important that point pressures be measured exactly at the line of cylinder diameter. This measurement is accomplished by limiting the travel of the scale beams by stops *G*, which form electrical contacts. The allowable vertical travel between the contacts at the end of the scale beams *F* is 0.003 in. With a 10:1 ratio of the beam arms, the movement of rollers *A* and *E* is but 0.0003 in. With this slight movement it is possible to measure the ring within close enough limits to determine the pressure on the cylinder wall at these points, regardless of the characteristics of the ring when confined in the cylinder. The scale beams *F* are graduated in pounds and tenths of pounds. The poises *H* slide along them and are set at the low limits specified for ring-point pressures. The ring is then inserted, and the results are read through the PLUS and MINUS windows below the scale and beneath the rollers. If the PLUS lights show, this indicates that the ring is in excess of the low-limit pressure specified. If either of the MINUS lights shows, the ring is rejected for insufficient point pressure. If no lights show, this indicates that the ring has a pressure on either point equivalent to the low limit specified. This apparatus has proved to be sufficiently fast so that it can be used for quantity checking. The scale is adjustable to receive rings from 2½ to 6 in. in diameter and up to ½ in. wide.

Piston-Ring Shape Analyzed

Free piston-ring shape is most important in obtaining good performance and long life. The old method of weighing rings to determine their tension, or pressure, by applying sufficient load to close the joint on a diameter at 90 deg. to the diameter through the joint, indicates little or nothing. It results only in a comparison of the pressure required to deflect the back half of the ring to close the joint. This portion is of little importance and is not difficult to form.

The most important portion of the ring to form is the 180 deg. opposite the back half; that portion having the joint. The ring must be shaped to exert maximum pressure on the cylinder wall at the points, and yet have a shape on either side of the joint to exert sufficient pressure immediately after installation to prevent blow-by. To accomplish this result has required considerable study of the proportions of radial wall thickness and free-joint opening to diameter. When these proportions are correct, the ring will conform at once to the curve of the cylinder wall with sufficient wall pressure at all points.

Fig. 3 shows two extremes in ring shape, exaggerated, in the roller gage shown as Fig. 1. The dotted ring in Fig. 3, shown with the points lying away from the rollers at the joint, would exert no pressure on the cylinder at the points. The ring shown in contact with the rollers at the points is of such shape that there is no contact on the next three rollers on either side of the joint, although the point pressures may be sufficient. When these two rings are confined in an endless flexible band and checked for what is commonly known as "circularity", the former ring will be "flat"; in other words, the diameter across the joint will be less than the diameter at 90 deg. to it. When confined, the latter ring would show "long," or the opposite. Rings that are "long" would indicate that the point pressure is high, and rings that are "flat" would indicate that the points were low. From this it can be seen that it is impossible to determine accurately the shape characteristics of a ring by the flexible-binder test alone. After correct shape has been accomplished, this sort of test can be used to some advantage.

Effect of Incorrect Piston-Ring Shape

Comparing the performance of the rings shown in Fig. 3, the rings having no point pressure allow excessive blow-by at moderate speeds and give fair oil economy at low speed. The rings having point pressure resist blow-by up to a higher speed; however, the portions lying away from the rollers permit the excessive passage of oil. At speeds in excess of 3200 r.p.m., the pressures from the combustion chamber will collapse the ring lying away from the rollers on either side of the joint,

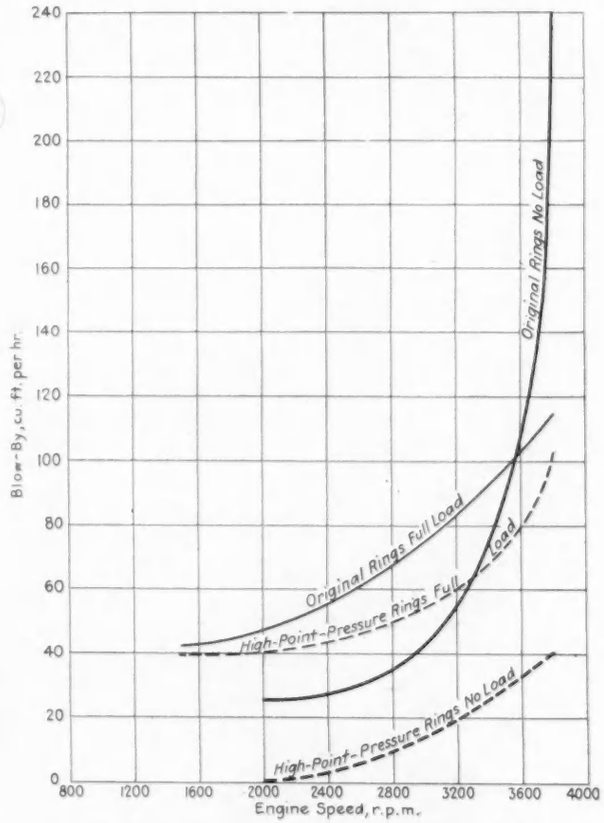


FIG. 4—CHART SHOWING THE BLOW-BY RESULTS OF A CONVENTIONAL SIX-CYLINDER HIGH-SPEED ENGINE AFTER 24,000 MILES OF OPERATION ON THE ROAD

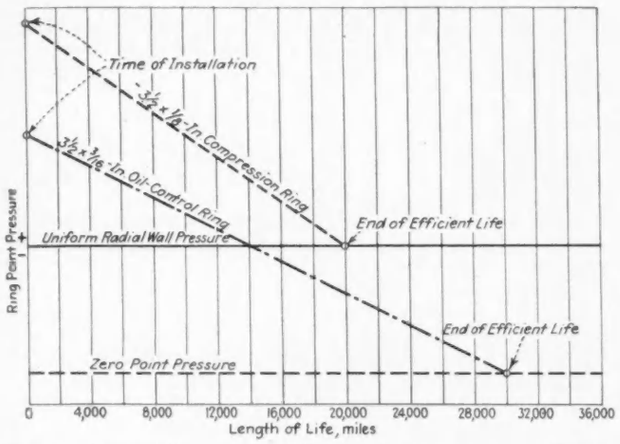


FIG. 5—CHART DEVISED TO SHOW DIFFERENCES IN SHAPE AND RADIAL-PRESSURE CHARACTERISTICS, AND THE RELATION TO THE LIFE OF BOTH OIL-CONTROL AND COMPRESSION RINGS

but not as readily as they will the ring lying away from the rollers directly at the joint.

The most critical speed at which blow-by occurs is at the moment when the combustion-chamber pressure changes from maximum to minimum; that is, when passing from open throttle to closed throttle. So long as the throttle is open, provided the cylinders and pistons are in good condition, the points of the ring are held on the cylinder wall during the compression, power and exhaust strokes by a component of the pressures exerted behind them. The instant the combustion-chamber pressure is reduced so that it is insufficient to hold the rings on the cylinder wall, the ends begin to vibrate and maximum blow-by occurs. This burns the protective oil film from between the cylinder and ring faces, which action usually takes place unawares. However, it can be measured accurately by laboratory tests.

Fig. 4 shows the blow-by results of a conventional six-cylinder high-speed engine after 24,000 miles of operation on the road. The engine was mounted on a dynamometer and first checked for blow-by with the original rings, which had insufficient point pressure. It should be noted that the maximum-power blow-by at 3800 r.p.m. is 115 cu. ft. per hr., whereas the maximum blow-by while idling, with the same equipment, is 240 cu. ft. per hr. at 3800 r.p.m.

The fact of prime importance in connection with blow-by while idling is that the rings left the cylinder wall and began to "vibrate" at 2600 r.p.m., and the blow-by increased at a high rate with engine speed. The original equipment was then replaced with correctly shaped rings having sufficient point pressure. The results after installing the high-point-pressure rings plainly indicate that the maximum blow-by occurs when combustion-chamber pressures are low, so that the action of the piston, as well as the distortion and irregular wear of the cylinder, affect the functioning of the rings most. Without question lubrication has considerable to do with the way in which rings function. However, the major requirement is sufficient radial point pressure, together with proper conformity of the ring to the cylinder wall at all speeds. A piston ring is no more than a spring, and, if the spring pressure is insufficient to hold it on the cylinder wall, ring vibration at the critical speed is bound to occur, with all of its attendant bad effects.

Length of Life of Rings

Fig. 5 represents an endeavor to devise a graph to show differences in shape and radial-pressure characteristics, and their relation to the length of life of both oil-control and compression rings. For maximum life, compression rings must start with a maximum point pressure; at the same time they must contact with the 17 rollers in the roller gage. As the concentric ring wears and the joint opens, the point pressure reduces at a very rapid rate. On the best shaped 3½-in. compression ring, these pressures reduce at a rate of 0.23 lb. per 0.001 in. of radial wear. It has been determined that a ring of such shape will continue to perform satisfactorily until approximately 0.010 in. is worn from the face, which corresponds to approximately 20,000 miles of normal operation.

An ideal oil-control ring obviously should have uniform radial wall pressure. This can be accomplished initially; but it cannot be maintained, due to wear. Therefore it is necessary to design this ring so that it will pass through a period of efficient life plus and minus of the uniform-pressure line. It will be noted in Fig. 5 that the point pressure is somewhat lower at the time of installation, when compared with the compression ring. It should be noted how the efficient-life line passes through a zone above and below the uniform-pressure line until it reaches zero point pressure, representing the end of the efficient life of the oil-control ring. In developing the oil-control ring it has been possible to double the radial-wear factor, which has increased the efficient life to approximately 30,000 miles. It is then obvious that an oil-control ring so designed is at a point of maximum efficiency when half worn out.

Fig. 6 is a chart showing 3½ x ⅛-in. compression rings of five different makes. They have been graded in relation to their efficient life; namely, Groups I, II, III, IV and V, of which Group I represents the ring having the longest life. The values shown at the right represent actual radial wall pressure in pounds. The values at the left represent wall pressures in pounds measured by the radial-pressure scale. Because the scale confines the ring within five rollers, it is necessary to divide these values by 1.3 to determine the actual point pressure exerted on the cylinder wall by rings correctly shaped. This factor does not apply to all diameters.

It should be noted that the dotted line specifies a low limit of 3½ lb. when measured by the scale. It is the opinion of the authors that such rings should show a pressure in excess of the minimum shown, if maximum life is to be assured. The ring in Group V has the shortest life and is more or less worn out before it is put into operation. With a known pressure drop of 0.23 lb. per 0.001 in. of radial wear, this ring would reach the zero pressure line after approximately 0.003 in. wear, or, in other words, almost before worn to a

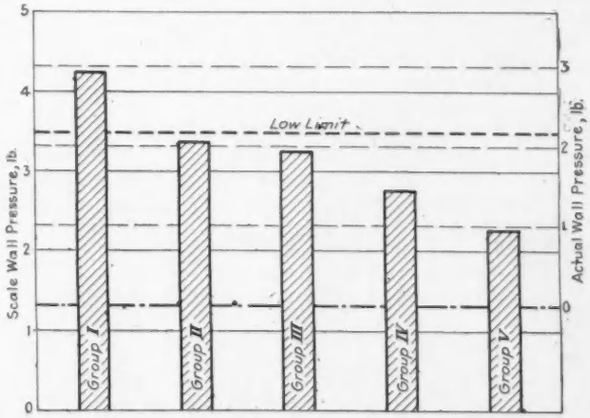


FIG. 6—RESULTANT POINT PRESSURES OF FIVE MAKES OF 3½ x ⅛-IN. COMPRESSION RING WHEN MEASURED BY RADIAL-PRESSURE SCALE AND BY ROLLER GAGE

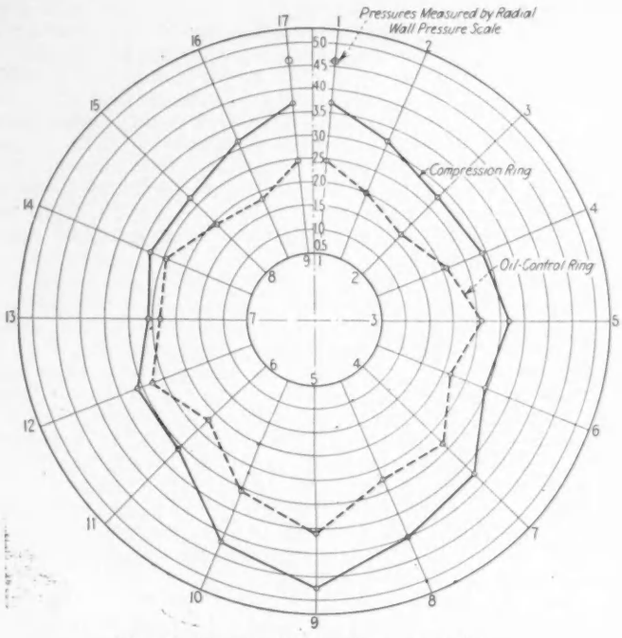


FIG. 7—RADIAL WALL-PRESSURE CHART

The Roller Gage Illustrated in Fig. 1 Has 17 Radial Holes So That a Plunger Can Be Inserted near Each Roller. The Gage Is Set Up in an Accurate Scale, and Sufficient Pressure Is Exerted on the Plunger To Release the Adjacent Roller. The Resultant Pressure Can Then Be Recorded and Plotted on a Radial Wall-Pressure Chart in the Way Here Indicated

satisfactory face bearing. The other rings shown have a life proportionate to their respective comparative point pressures. This same reasoning can be applied to the oil-control rings.

Plotting Radial Pressure

The laboratory roller gage has 17 radially drilled holes, so that a plunger can be inserted at each roller. The gage is set up in an accurate scale, and pressure exerted on the plungers sufficient to release the roller, as shown at *A* in Fig. 1. Resultant pressures can then be recorded and plotted on a radial wall-pressure chart, as shown in Fig. 7. The high pressure on the diameter through the joint is necessary to accomplish long life, as explained for Fig. 5. The values may seem to be somewhat irregular, but it must be realized that a considerable difference in pressure will result from variations of tenths of thousandths of an inch in face contour. In other words, if the rollers have a variation of more than 0.00005 in., it is impossible to draw an accurate conclusion with respect to resultant radial pressures on new rings that have not been rubbed to a full face bearing. The rings that were plotted and shown in this chart were new rings that had never been operated in a cylinder.

Engine Performance in the Field

Referring again to Fig. 6 in connection with engine performance, and the effect when fitted with rings in Group I as compared with those in Group V, the rings in Group I have an efficient life of approximately 20,000 miles. The rings in Group V have insufficient point pressure from the very beginning, which results in ring vibration that causes excessive blow-by and a fast rate of wear. It is doubtful if rings in Group V have an efficient life of even 5000 miles. However, thousands of engines have been built with both types of ring, and, from an accurate field survey, we find the first ring change is made after about 18,000 miles, regardless of the type or performance of the original rings. With these facts before us, and knowing that the rings in Group I have an efficient life of approximately 20,000 miles, it means that thousands of motor-cars are being driven four times the efficient life of the original ring equipment. We believe that engine life could be increased from 25 to 50 per cent, still continuing to make the ring change at 18,000 miles, if rings were used having correct characteristics, as in Group I.

If compression rings are changed before the end of their efficient life, oil-control rings will then show a greater life than the best compression rings, as indicated in Fig. 5; however, oil-control rings have a very short life when used in combination with poor compression rings. Excessive blow-by causes undue wear to cylinders, pistons and rings and shortens the life of an engine considerably. For this reason it is imperative that all rings be replaced periodically.

Effect of Cylinder Finish

The smoothness of cylinder finish materially affects piston-ring life. Cases have been known in which rings have worn out completely in 500 miles, after having been installed in roughly finished cylinders. It is extremely important that cylinders be finished as smoothly as possible, to assure maximum ring life. The whole problem is: How rough is smooth? We have developed an instrument to determine the smoothness of cylinder finish. With such an instrument, ring life can be more or less predetermined. This instrument is placed on top of the cylinder to be checked. It has a graphite pencil lead that presses against the cylinder wall with a constant pressure. When it is moved up and down a definite number of strokes, the wear of the graphite pointer, as indicated on a dial on top, denotes or grades the smoothness of the surface. This instrument should assist materially in eliminating the human element for determining the smoothness of cylinder finish.

Air-Transport Maintenance Problems from the Service Viewpoint

Aeronautic Meeting Paper

By Luther Harris[1]

VIEWS of the maintenance chiefs of all major air-transport lines, based upon their experiences in this field and as transmitted by them through the Maintenance Committee of the Aeronautical Chamber of Commerce of America, Inc., are embodied in the paper. Representing as it does the collective experience of the best minds in the field, the paper is particularly significant and worthy of the consideration of manufacturers, engineers and others directly concerned with the problems presented.

As to fuselage and wing coverings, it is stated that fabric has a definite advantage when considering weight and emergency repairs. Airplanes covered with fabric can be restored to service quickly in cases where, with the same damage, replacement of other forms of covering would cause the plane to be laid up in the shop at a time when it is most needed. With the new improved finishes, fabric is said to be as satisfactory as any covering available.

Other subjects treated include landing-gears and wheels, controls, features of cabin interiors, gasoline tanks, fuel systems and oil tanks, engines, electrical equipment and other accessories. The author states that every consideration should be given to accessibility.

Discussers state that the air-transport operator is now in a position to specify to the manufacturer various requirements that must be met before equipment will be bought.

MOST maintenance problems of air-transport operation consist of conventional troubles which have been found in the early stages of other forms of transportation. Since the air-transport industry involves the application of power in a manner different from that for any other form of transportation, it is necessary to save weight to increase power efficiency. Other forms of transportation can apply 1 hp. of energy in the form of horizontal drawbar pull and move a 33,000-lb. mass several feet in 1 min. by using present-day bearings. In the air-transport industry, we can expect to move this mass only 1 ft.

Naturally, problems involved in maintaining an airplane fleet are looked upon by service departments from an entirely different viewpoint from that of designers, because, in other branches of the automotive industry, we have always made any mechanism heavier after it broke; but, in most cases, this is the wrong thing to do in the aircraft industry. The designer must use and continue to broaden his experience to strike a medium between weight, strength and accessibility. The Aeronautical Chamber of Commerce is not an exclusive engineering group but, fortunately, it is in a position to offer certain data in connection with those problems that are vital to designers and engineering groups. Some of these problems are explained as follows:

Fuselage and Wing Covering

Fabric covering has a definite advantage when considering weight and emergency repairs. In many cases an airplane can be restored to service where the same damage to other kinds of covering would necessitate shop repair when the plane was most needed. In combination with the new improved finishes, fabric is as satisfactory as any covering available; however, it constitutes a greater fire hazard.

Corrugated duralumin gives maximum strength with minimum weight, but the thickness is such that it is difficult to prevent breakage at points of vibration, especially near rivets. It has proved more economical in the past. In fact, operators maintaining fabric-covered, plywood-covered, and corrugated-duralumin-covered wings on a single division of their lines, where maintenance costs could be studied carefully, report a marked advantage in favor of the corrugated-duralumin covering. Inspection, however, is more difficult and it is impossible to detect crystallization. Sheet duralumin formed to the fuselage has definite advantages over both the other types, because a much thicker duralumin can be used. Its weight is perhaps its greatest disadvantage.

Landing-Gears and Wheels

Inaccessibility has caused some of the greatest trouble in the past. Since landing-gears have evolved from the spreader-bar type, they have in many cases been considered a part of the fuselage. The air-and-oil type of strut has proved very satisfactory, and in most cases the installation is very accessible.

High-pressure-tired wheels have always been the choice of the mechanical personnel; however, the present semi-low and low-pressure-tired wheels are being made accessible. Replaceable bushings in V-shaped structures and similar mountings must be designed into future landing-gears. Not enough attention has been paid to assure ample-sized bearing areas on all moving parts. Bushings should be easily removable and acccessible for reaming. The replacement of common bushings in wheels by tapered frictionless bearings is a decided improvement. Roller and ball bearings increase take-off ability and lengthen bearing life. Tail wheels and forks mounted on the rudder and fin post should have replaceable metal inspection doors over the last station of the fuselage whenever practicable, as failure at this location must be expected.

Brake troubles are now much fewer, but failures are still too frequent. Too much attention must still be given to brakes to keep them in satisfactory condition for safe operation. It is evident that braking surfaces are too small for their required duties. Hence it is necessary that brakes be nearly perfect to give satisfactory results. Slight wear, or having water splashed on them, results in reducing the braking force to less than that required by the pilot. Brakes having a greater

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margin of safety are required so that they will not need readjustment after every trip.

Controls and Control-Surface Mounting

Rigid push-and-pull controls must have bushings at all joints to facilitate economical service. It is also suggested that they be double-safetied like Scintilla Magneto Co. control rods. Pulleys for control wires are too small in many cases. Ball-bearing pulleys have shown great savings in service. Control-surface hinges have insufficient bearing surface and in most cases are not properly bushed. It is suggested that rigid push-and-pull control rods be made of harder metal than that of the guides; otherwise, the guides will cut the rods.

Cabin Interior Features

Suitable floor covering has always been a problem. Its weight and appearance are important, but as air-sickness necessitates keeping floor covering clean and odorless, no floor corners that cannot be cleaned and washed quickly are permissible. Some operators have found plywood covered with special airplane linoleum satisfactory for cabin floors. The real difficulty is decay of the plywood due to moisture from the cleaning water. A method for waterproofing plywood or sealing the edges so as to prevent it from absorbing moisture would be a great money saver.

Fabric or aeroboard interiors with lacquer finish stain too easily, especially if the cabin is decorated in delicate colors. A Bakelite interior trim in natural color without coating of lacquer would reduce maintenance expense. Conventional sockets for cabin lights have given considerable trouble when installed where vibration is great.

The cabin interior should be free from unnecessary fittings and fancy trimmings. Steel chairs provided with removable cushions and covers are easily maintained and keep their good appearance a long time. Suitable receptacles and fittings should be provided for use by passengers in cases of airsickness.

Ventilation and heating still constitute a problem that must be solved. Ventilation controlled by the pilot seems to have definite advantages. The pilot should be provided with an indicator showing the temperature of the cabin, so that he can suitably regulate the heat. It is suggested that an ordinary ice-warning indicator be used, and that the actuating ball or bulb be located inside the passenger cabin. Hat racks, foot controls for heaters and other objects that passengers can stumble over or bump their heads against have been a source of trouble and complaints. All of these are expensive to maintain and to redesign.

Tanks and Fuel Systems

Until better dependence can be placed upon gasoline and oil tanks, every possible thought should be given to providing accessible installation and foolproof mountings. The time required to make emergency changes has always been too great. On some types of airplane it has been difficult to fill and to drain gasoline and oil tanks. All need sumps. If the filler caps of gasoline tanks were mounted on a large manhole plate equipped with gaskets and fastened by anchor nuts, much service work would be eliminated in cases of corrosion or when it is necessary to clean the interior of tanks. In many cases fuel strainers should be placed in a more accessible location; in some gasoline tanks they are not removable, and brass or bronze eventually will corrode; they should be removable. Fuel lines should be installed so that they can be removed readily for annealing. The use of a color code for the pipes on the larger planes, so that they can be quickly and easily traced, is suggested.

Fuel tanks of oval shape and welded aluminum construction are almost trouble free when properly made. Flat or rectangular-shaped tanks are trouble makers, regardless of whether they are made of brass or of duralumin. Special attention given to the way the nipples for line connections are attached to the tank will save many hours of maintenance. In place of connecting these nipples directly to the skin of the tank, if the tank is designed so as to include a sump of much heavier material than the rest of the tank, the nipples can then be anchored to the sump. This will eliminate leakage caused by vibration, which cracks the metal near the base. When baffle plates are riveted to the outside face of gasoline and oil tanks, they should protrude through and be welded to the wall of the tank. In service, rivets often break and lodge in the sediment well of the tanks, or, occasionally, if no screens are in the line, the rivet heads will find their way into the gasoline feed lines.

Engines and Accessories

After it has been put into service, very little can be done to an engine to reduce the maintenance cost. The arrangement of accessories and the installation work govern the cost to a great extent. Of these, perhaps correct installation saves more money. Ignition systems, carbureters and oil-pumps are the accessories that require most frequent inspection. Unless the accessibility designed into the engine and engine nacelles by the manufacturer is maintained in the installation, the operator will pay accordingly.

Attention is called to the haphazard way in which many manufacturers install engines. In many cases there apparently was no prearranged plan as to how the gasoline lines should be run, carbureter-control rods attached, fire extinguishers installed, electric wires enclosed and the like. The result is that the mechanic who installs the engine at the factory is largely left to his own devices. After looking at the tangle of copper tubes, control rods and wires that one usually sees behind the engine, it is evident that no great amount of engineering thought was devoted to remedying this detrimental feature. In the design of engine controls a lack of ruggedness to enable them to withstand the constant vibration to which they are subject is evident. Further, the installation should be such that easy access to strainers and adjustments is maintained.

Generators, Starters and Batteries

The amount of electrical energy needed for large transport airplanes has increased so greatly that it is now difficult to maintain these requirements. In some of the larger planes, cabin, instruments and navigation lights cause a constant drain of 60 amp. from the battery. In most cases the current necessary for radio-filament heating and dynamotor is 60 amp. or more; for landing lights, 70 amp.; and for starters, at least 110 amp. Even though all these are not used at the same time, all we can hope to do is to balance the fixed drain with a generator and charge the battery at a conventional rate. The larger batteries in use will permit a discharge rate of only approximately 12 amp. for 5 hr, or 70 amp. for 25 min. On this basis it is necessary to charge the battery after three or four night landings or if radio is used for a period of 30 min.

If the demand for electrical energy continues to grow, other steps must be taken or maintenance costs for this item will become prohibitive. Batteries with the non-spill feature are desirable for military service, but, on transport planes for airline operation, batteries are needed that will carry heavier loads without increasing the battery weight and which have a non-slosh rather than a non-spill feature.

Every operator is now faced with the problem of

trying to use one small battery to do the work of three or four. It should be the duty of the manufacturers to furnish with their radio sets generators which will eliminate the use of a battery, except, probably, for emergency use, as the lights on the plane require about all the current one battery should furnish. The radio manufacturer also should give the operator some constructive help in the shielding of ignition. It is hoped that we shall soon be able to influence the manufacturers of radio equipment, spark-plugs and magnetos to cooperate in developing a system of shielding that will lessen the burden of the operator. We have reached a point where the operator will have to demand this before he buys radio equipment.

Difficulties Caused by Radio Equipment

The maintenance departments of the various airlines consider radio equipment very essential when operating under adverse weather conditions; but, with present equipment, there will continue to be delayed departures and ignition troubles, usually caused by ignition shielding. Condensation in shielded-type spark-plugs causes a considerable amount of this trouble. However, trouble can be expected to develop at any place between the distributor block and the spark-plug, because the entire high-tension circuit is covered by a ground connection which lies ready to show up any insulation weakness or moisture at connections. The solving of these problems should be the responsibility of both the radio-instrument and the accessory manufacturers.

Suitable Tools and Equipment Necessary

The cost and speed of maintenance and line service depend largely upon the type of tools and equipment available for the particular kind of aircraft. Most manufacturers are not familiar with the needs of airline work, but during the last two years the airline operators have developed special service equipment to meet their needs, comprising handling equipment, workstands, jacks and tools for various kinds of special work. Heretofore the lack of this kind of equipment has been the source of trouble and expense. Cooperation between airlines and manufacturers will lead eventually to standardization of this equipment and reduce the operating cost.

THE DISCUSSION

W. E. BERCHTOLD[2]:—The present outlook is much more hopeful than at any time since air-transport lines began operating, because conditions have changed considerably. Several years ago the air-transport operator had to accept whatever equipment was available and provide the best service possible, but he is now in a position to specify to the manufacturer various requirements that must be met before the operator will buy equipment. It should be said that all manufacturers are trying to do better regarding inclusion of maintenance items when designing new airplanes.

RICHARD M. MOCK[3]:—The regulations of the German Government for aircraft specify, I believe, a definite reduction in calculated allowable bearing strength in moving parts such as the bearings in landing-gears that occur at the bolted or hinged fittings. The moving surface suffers the reduction and the fixed surface is allowed the full bearing strength for static bearing. Consequently, this requires that moving surfaces must have greater area for the same bearing load. For example, in a forked fitting having a tongue between the legs of the fork and fixed by a bolt through the legs and the tongue, where the tongue rotates as would a landing-gear member, the bearing area of the tongue-bolt surface would be greater than the total of the two legs-bolt areas if the bolt were fixed relative to the two legs and the tongue rotated on it. My recollection is that moving bearing surfaces are allowed only one-third the bearing strength allowed to static or fixed bearings.

CHAIRMAN R. C. MARSHALL[4]:—I know from experience that the bearing problem for the controls is one of the most difficult problems we have had. In many instances the manufacturers seem to feel that plain bearings—as small as possible and using as few of them as possible—will reduce maintenance costs; but, on the contrary, such bearings wear and not enough of their material is left to make it feasible to bush them. Sometimes a major replacement must be made merely to replace the bearing.

MR. BERCHTOLD:—Recent discussions in meetings of the Chamber by representatives of the various maintenance departments have brought out the fact that bearing surfaces have been made entirely too small. Anything that is done to enlarge them will be a great aid to maintenance departments.

QUESTION:—Which is the larger item, airplane or engine maintenance?

MR. BERCHTOLD:—My impression is that the maintenance cost for the airplane is higher than that for the engine.

CHAIRMAN MARSHALL:—It would depend upon what was included as maintenance cost. For a 300-hp. single-engine airplane, the maintenance cost for the plane would be about equal to that for the engine. If the periodic overhauls of both the plane and the engine are included, the maintenance cost of the engine, which requires more frequent overhauls but costs less per overhaul, probably will equal the total cost of the plane overhaul. But I think the engine maintenance is greater than the maintenance on the airplane for a multi-engine plane, because the airplane usually will last for two years, with line maintenance, before it needs complete overhaul; whereas the engines need a complete overhaul frequently. I refer to the conventional fabric-covered plane. Our experience with the plywood-covered plane is that it does not require covering any more frequently than does the plane covered with fabric; but the cost of recovering with plywood is much higher than that for fabric. Engine maintenance for an airplane equipped with three engines is more than twice that for the airplane.

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[3] Jun. M. S. A. E.—Assistant to Mr. Fokker, A. H. G. Fokker, New York City.

[4] President, Trans-American Airlines, Detroit.

Brake-Drum Metallurgy

By F. L. Main[1]

Semi-Annual Meeting Paper

EITHER steel or cast iron will provide a good braking surface provided the grain structure is laminated pearlite, according to the author. Such a structure can be secured in pressed steel by alloying or by case-hardening, in high-carbon steel rings welded to a stamped back and in centrifugally cast iron by careful control without alloying. Uniformity of analysis is important and control of the rate of cooling is still more important in castings. The graphite content of iron is not considered important as a lubricant. Methods of centrifugal casting and of testing are illustrated; also the form and microstructure of representative brake-drums.

Discussers agree as to the microstructure needed and present additional views as to ways of securing that structure and the desirability of capacity for absorbing and dissipating heat. They believe grain size and strength more important than hardness.

FREE-WHEELING and higher road speeds have introduced no new brake-drum difficulties but they have intensified troubles that have been with us for years. Heat, chatter, distortion, fading and washout are related to matters of brake-drum design such as size, weight, proportion and contour; but scoring, the chief difficulty arising from higher speeds and faster stops, is primarily a problem of metallurgy.

Intensive research has shown that scoring can be prevented in drums of the designs now in production by securing a grain structure of laminated pearlite at the braking surface, if present speeds, pressures and temperatures are not exceeded.

The test equipment used for arriving at these conclusions includes the tire-driven brake dynamometer shown in Fig. 1, which can be operated continuously under conditions simulating various car speeds and weights. Attached to this dynamometer is a Bristol recording tachometer for taking deceleration curves like those in Fig. 2 of brakes employing drums of different kinds. The machine shown in Fig. 3 is used for testing drum scoring. It is operated automatically at selected speeds and loads, with water and dirt sprayed around the brake-drum every sixth cycle.

[1] A.S.A.E.—Sales engineer, Kelsey-Hayes Wheel Corp., Detroit.

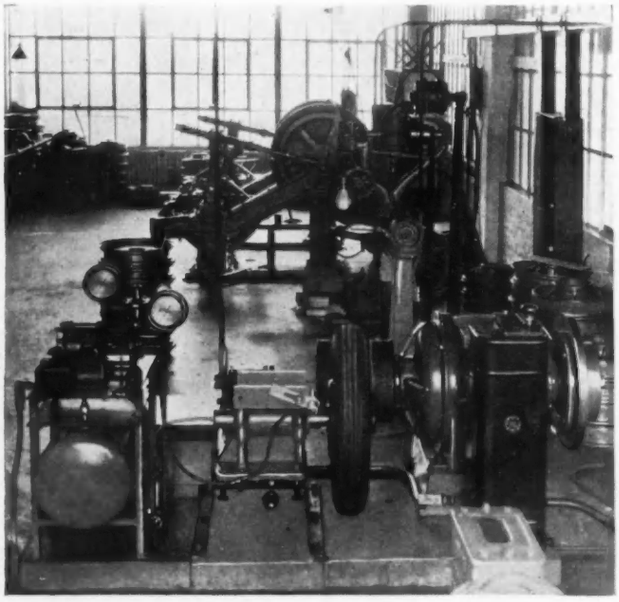


FIG. 1—TIRE-DRIVEN DYNAMOMETER FOR BRAKE TESTS

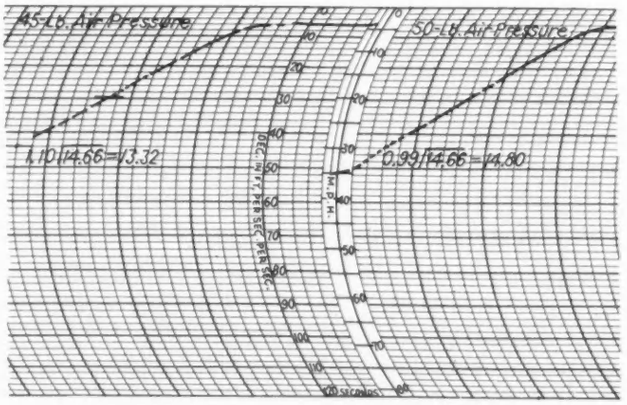


FIG. 2—TACHOMETER RECORD FROM BRAKE DYNAMOMETER
The Deceleration in Feet per Second per Second Is Calculated on the Strip for Each Run Shown

As an example of stamped drums tested, we shall consider the one in Fig. 4, made of S.A.E. 1025 stock. A photomicrograph at 100 diameters (Fig. 5) shows the characteristic cold-working bands in the braking surface. The white is ferrite, the black spots are pearlite, and the fine black lines are the boundaries of the ferrite grains.

Under braking conditions with temperatures from 700 to 1000 deg. fahr., as indicated by thermocouples, the whole piece of the material tends to revert to its original shape because of the unreleased internal strains. The braking flange goes out of round and creates areas of high pressure, where the soft ferrite is picked up on the brake lining, and scoring follows.

Fig. 6 shows the slight scoring and general wear of the braking surface on a piece of the same brake-drum after it had been operated for 500 stops from 37.5 m.p.h. on the drum-scoring machine. The tendency to go out of round at braking temperatures can be reduced by normalizing; but this has the disadvantage of lowering the surface hardness, so that the drum may score at lower pressures than without normalizing. If the drum is quenched, to avoid this difficulty, holding the drum to shape and to size becomes a serious problem.

Alloying or Carburizing Prevents Scoring

Alloy steel has been suggested for stamped drums. High-carbon steel proved unsatisfactory, because it was drawn hot and was difficult to keep within commercial limits after forming. High-manganese steel, containing 1.75 per cent of manganese and 0.30 per cent of carbon, also has been tested in brake-drums. The material was easily drawn, stood up well under test, and offers improved wearing qualities for those who desire to continue to use stamped drums. Fig. 7, which is a photomicrograph of this steel at a magnification of 200 diameters, shows some cold-working bands in a vertical position. The ferrite (white) is very well broken up; the black is pearlite.

Carburizing is another method of supplying a satisfactory braking surface on a stamped drum. A conventional stamped-steel drum of S.A.E. 1010 steel was carburized to a depth of 0.020 in., cooled in the pot and rolled on the surface. Two results were accomplished: the temperature of carburizing eliminated the working strains and the increase of the carbon on the braking surface provided a skin hard enough to resist scoring. The drum was taken off test after 5000 stops without deformation or scoring. The lining used readily scored ordinary pressed-steel drums. The coefficient of friction was 0.4.

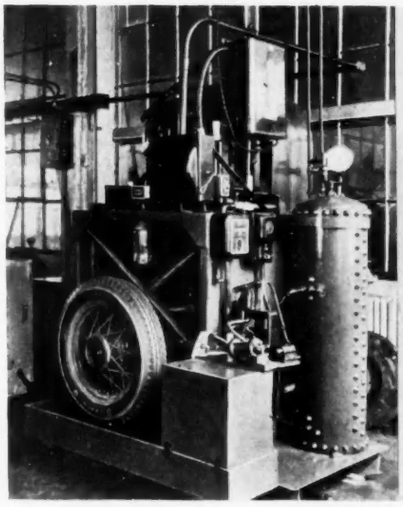


FIG. 3—TESTING MACHINE FOR DRUM SCORING

The Cover Is Placed over the Wheel While It Is Running. Automatic Control Causes Successive Stops from Selected Speeds. Water and Dirt Are Sprayed on the Brake during Every Sixth Stop

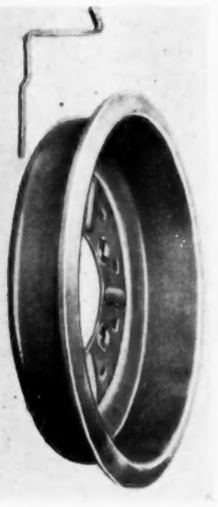


FIG. 4—CONVENTIONAL STAMPED-STEEL BRAKE-DRUM

Fig. 8, which is a photomicrograph at 500 diameters of the braking surface of this drum, shows the carburizing. The white is free cementite. In places the finely laminated pearlite tends to merge into sorbetic pearlite. The carbon content is about 1 per cent. Tests indicate that carburizing should prove satisfactory for those who wish to use a stamped drum because they consider the saving in weight to be fully as important as the first cost.

Quenched cyanided brake-drums also were tested. While the performance was fully as satisfactory as that obtained with a carburized drum, the percentage of warping was high and it was difficult to hold the drums true to shape after quenching.

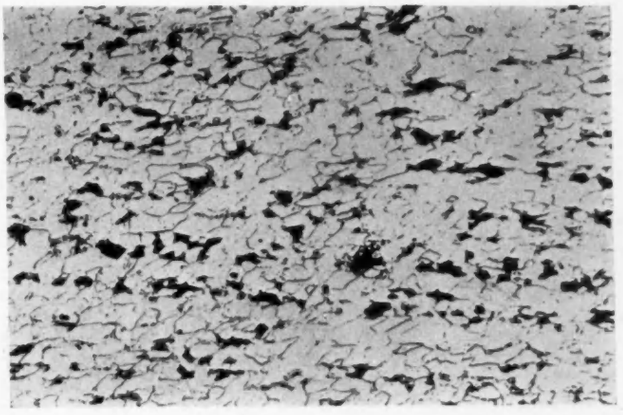


FIG. 5—PHOTOMICROGRAPH OF BRAKING SURFACE OF DRUM SHOWN IN FIG. 4

The Material Is S.A.E. 1025 Steel, and the Magnification Is 100 Diameters

Rolled-Steel Ribbed Drums

Ribbed steel drums are of two types: those in which the straight mill-section stock is rolled from strip, coiled and welded (Fig. 9), and those for which the ring is rolled endless from a hot billet (Fig. 10). A stamped back-plate (Fig. 11), which may be either butt welded or arc welded to the ring, is used in both designs. Experience indicates that arc welding is superior to butt welding, because of the difference in carbon content between the back and the ring. To secure satisfactory performance, it is necessary to have a liberal flange on the back-plate, which should be a press fit on the ring. The weld should be continuous, as shown in Fig. 12. Experiments conducted with loose fits and with spaced welds showed that these features tended to produce chatter, apparently because of some motion between the back-plate and the ring, which probably was due to changes in temperature as well as brake pressure.

A mill-section drum that has been coiled and butt welded clearly shows the butt-weld line on the ring, as in Fig. 12. The ring must be normalized before sizing, to reduce the hardness and obtain a satisfactory structure at the weld. Both mill-section and hot-rolled rings have been made of alloy and high-carbon steels. Heat-

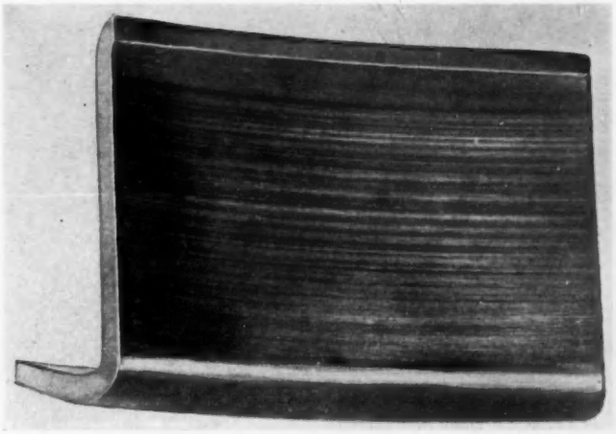


FIG. 6—SECTION OF THE DRUM OF FIG. 4 AFTER SCORING

This Drum Was Given 500 Stops on the Drum-Testing Machine from 37.5 M.P.H.

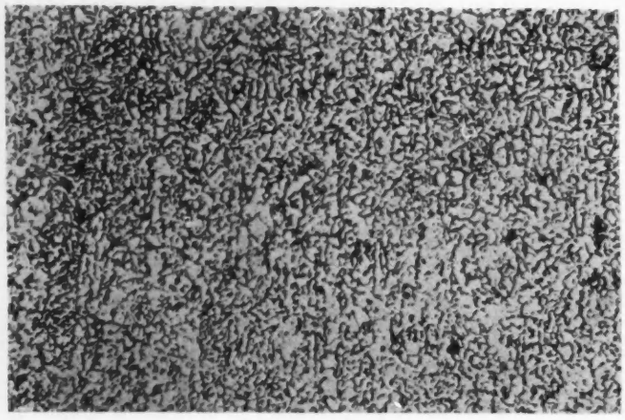


FIG. 7—PHOTOMICROGRAPH OF THE BRAKING SURFACE OF A HIGH-MANGANESE STAMPED-STEEL DRUM
Magnification 200 Diameters

treatment is employed to secure suitable grain structure on the braking surface of either type of ring.

The grain structure at 1000 diameters across the weld of a 0.74-per-cent-carbon mill-section ring (Fig. 13) is similar to and finer than that of the carburized drum, but it shows no free cementite. The light horizontal line is the center of the weld, and the structure is uniform throughout. The ring was normalized at 1700 deg. fahr. for ½ hr. and cooled in the air.

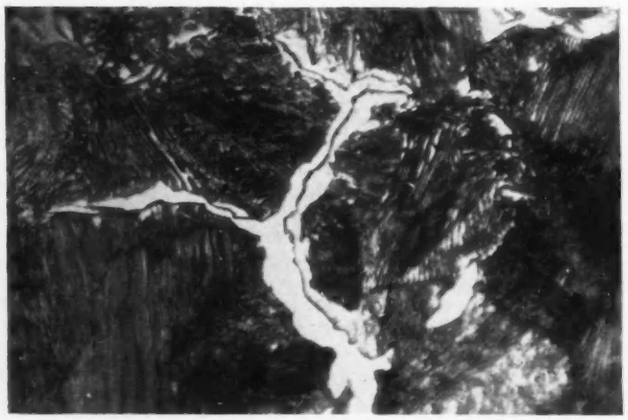
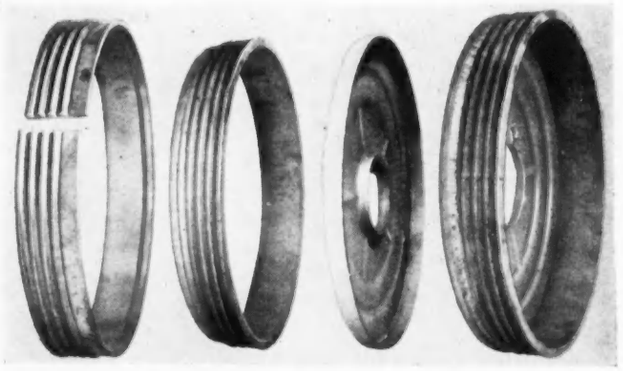


FIG. 8—PHOTOMICROGRAPH OF A CARBURIZED STAMPED DRUM
Magnification 500 Diameters

Ribbed rolled drums tend to vary in section. Care must be taken in sizing and in finishing the bores, so that this variation will not cause out-of-balance conditions which are conducive to chatter.

Cast Iron for Brake-Drums

Considerable thickness must be maintained at the back of a sand-cast iron drum to secure a strong casting. Full cast-iron drums tend to run out of balance unless machined all over, and this necessitates pouring extra stock in the back as well as in the flange. Cast iron of a desirable structure for a drum, as shown at 500 diameters in Fig. 14, has the same sort of laminated pearlite which gives such satisfactory performance in carburized stamped drums and heat-treated high-carbon rings. The heavy black line is graphite, the white is cementite and the fine black lines are ferrite.



BUILT-UP STEEL BRAKE-DRUMS
(From Left to Right) Fig. 9—Mill-Section Steel Coiled Ready for Welding. Fig. 10—Endless Rolled Steel Ring. Fig. 11—Back-Plate for Use with Either Type of Steel Ring. Fig. 12—Completely Welded Brake-Drum Comprising Ring from Fig. 9 and Back from Fig. 11

Malleable iron has been claimed to be desirable for brake-drums because of its high strength, but strength can be secured in gray-cast-iron drums by analysis and temperature control without sacrificing wearing qualities. In a photomicrograph at 100 diameters of the braking surface of a malleable-iron brake-drum (Fig. 15), we see the uniform distribution of the graphitic carbon in small quantities throughout the structure. This makes the ductility better than that of ordinary

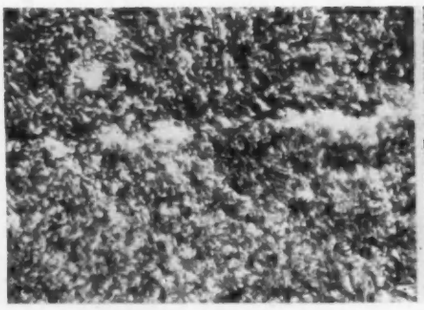


FIG. 13—PHOTOMICROGRAPH OF MILL-SECTION RING
Magnification 1000 Diameters. The Weld Is Indicated only by a Slight Variation in Grain Structure along a Horizontal Line. The Material Is 0.74-Per-Cent-Carbon Steel

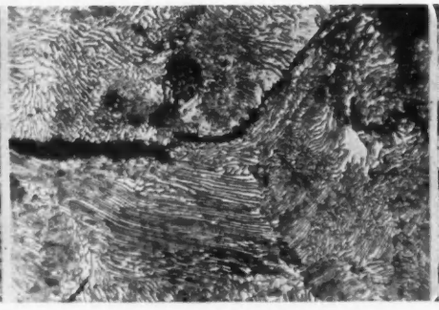


FIG. 14—DESIRABLE BRAKING SURFACE OF CAST IRON
Magnification 500 Diameters

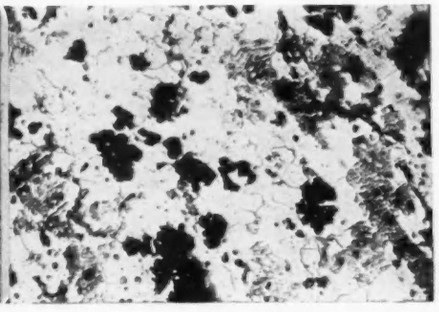


FIG. 15—BRAKING SURFACE OF MALLEABLE IRON
Magnification 100 Diameters

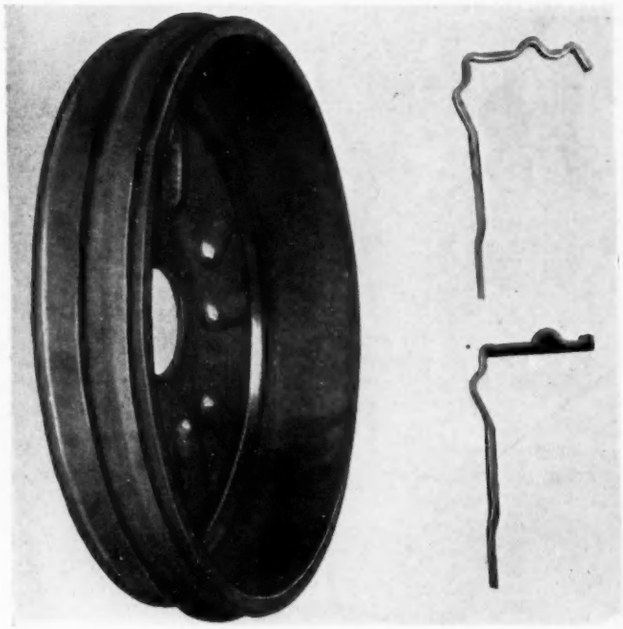


FIG. 16—A CENTRIFUGALLY CAST BRAKE-DRUM

The Braking Surface Has Been Machined. At the Right Are Shown Sections of the Drum before Lining and after Lining and Machining

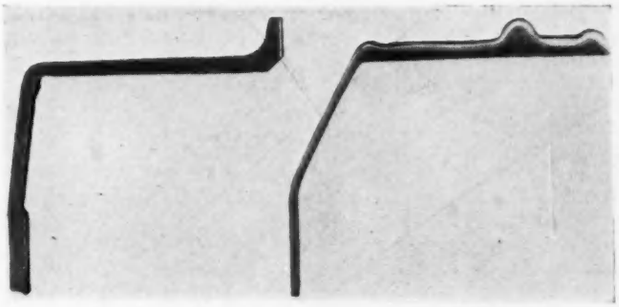


FIG. 17—SECTIONS OF THE CAST-IRON AND IRON-LINED STEEL BRAKE-DRUMS

Comparison of These Sections Indicates the Difference in Weight

cast iron, which has some value when hubs and drums are cast integrally. Some pearlite is to be seen.

Some of us have thought that cast iron is particularly desirable because of the lubricating qualities of the graphite, but graphite alone does not solve the problem. A brake-drum having a structure similar to that shown in Fig. 15, rich in graphite but low in pearlite, ran only 400 stops on the drum-testing machine before it was badly scored.

Cast Linings in Steel Drums

The ability of solid cast-iron drums to absorb, conduct and radiate heat is one of the chief arguments in their favor. Centrifugally cast drums (Fig. 16) were

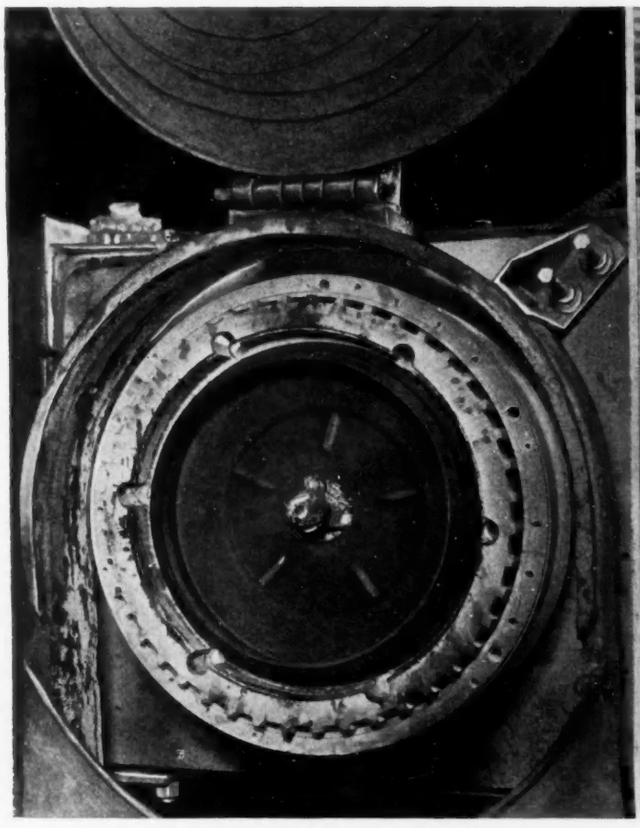


FIG. 18—CENTRIFUGAL CASTING MACHINE OPEN

The Red-Hot Drum Is Shown in Place, Supported Radially by a Ring That Is Toothed for Radiation and Held in Place by a Central Chuck

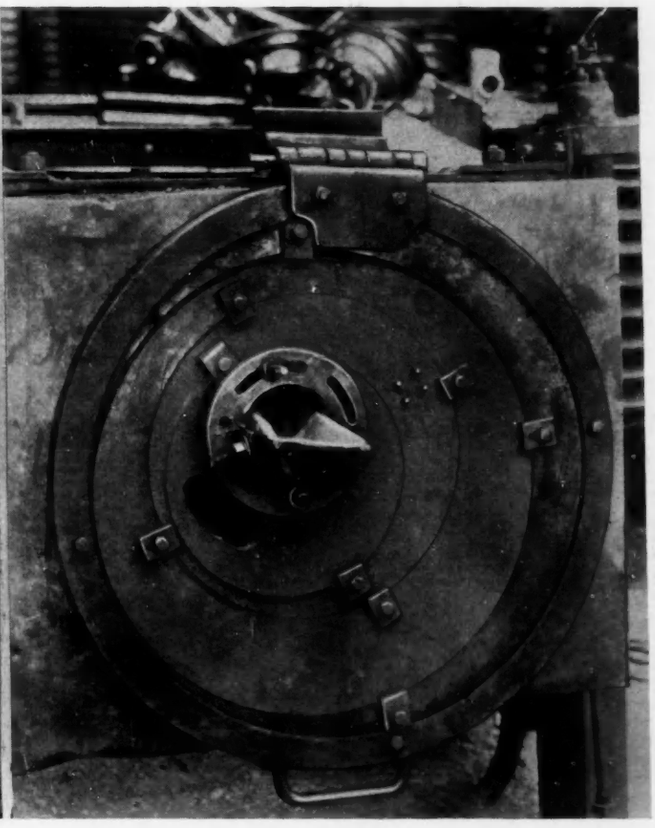


FIG. 19—CASTING MACHINE READY FOR POURING

Eccentrics in the Hinged Cover Provide Adjustment for the Pouring Position of the Spout

developed to provide this quality and at the same time to meet the demand for less unsprung weight. They comprise a steel outer shell—for lightness, strength, and uniform section—with a braking surface of iron centrifugally cast into the flange. No special heat-treatment is necessary to provide strength in the casting, and no alloys are needed to provide satisfactory wear. The only precaution that need be taken is to regulate the analysis and the rate of cooling so that a full pearlitic structure is obtained.

The back is made continuous with the flange, to assure strength and conductivity where the two join. This was also found desirable in mill-section drums, to avoid chatter and to eliminate areas of high unit pressure which often resulted from localized heat in the braking flange.

Comparison of the sections of a full-cast drum and a centrifugally cast drum in Fig. 17 reveal the saving in weight, which in some cases amounts to as much as 15 per cent. The thickness of the back varies for brakes of different kinds. Ribs should be used to provide the stiffest possible structure.

The machine for centrifugal casting, shown in Fig. 18 with an unlined drum in place, has a toothed ring to provide radiation. The pouring temperature is about 2800 deg. fahr., and the speeds for different drum diameters are 950 to 1100 r.p.m. The drum, heated red hot, is clamped at the center by a chunk and the outside is supported by the ring, as clamping at the periphery would tend to deform it and produce irregularities in the cast section. Fig. 19 shows the cover closed as it is when the drum is poured. The spout is attached to eccentric plates, to provide adjustments for drums of different diameters and shapes.

Structure and Bond Are Good

Fig. 20, which is a 100-diameter photomicrograph of a cross-section of a centrifugally cast drum, brings out the bond between the steel, which is light, and the iron, which is dark. The steel is carburized for some distance above the bond line. The white lines are free cementite, showing that the carbon in this portion of the steel has been raised to about 1 per cent. The effectiveness of the bond is indicated by the fractures in the iron, as seen in Fig. 21, which shows a section that has been cut from a centrifugally cast drum and hammered flat.

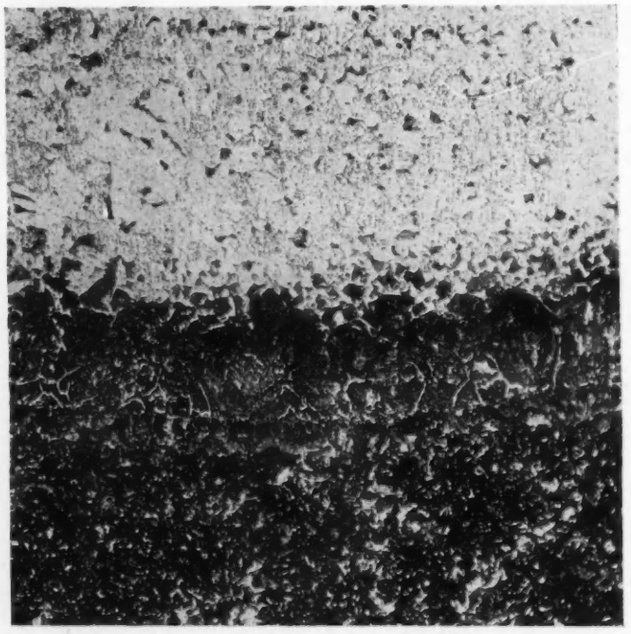


FIG. 20—SECTION SHOWING BOND BETWEEN SHELL AND LINING

A Fairly Distinct Horizontal Line Indicates the Joining between the Steel and Cast Iron. Carburization of the Steel Is Indicated by the Penetration of Dark Color above This Line. Magnification 100 Diameters

The scoring of a certain centrifugally cast drum after 1200 miles on road test shows that centrifugal casting alone is not a panacea for brake-drum troubles; correct composition is necessary. In Fig. 22, which is a portion of the surface of this scored drum at 100 diameters, we see the white ferrite, dark pearlite and black graphite lines, as well as some cementite. This drum should have been satisfactory, according to the analysis of the heat, but the structure clearly indicates that it was cooled too slowly for the silicon and carbon content.

A desirable structure of electric-furnace iron from a centrifugally cast drum which performed satisfactorily

FIG. 21—HAMMERED-OUT SECTION OF CENTRIFUGALLY CAST BRAKE-DRUM

The Tenacity of the Bond Is Clearly Indicated

in laboratory and road tests is shown at 500 diameters in Fig. 23. The large flakes are graphite, the white is cementite and the black is ferrite. The drum was taken off the testing machine after 1000 stops without scoring. Drums of the same character have been running on test cars for a long time with no signs of scoring. Analysis of the iron shows from 2.2 to 2.3 per cent of silicon and 0.20 to 0.25 per cent of nickel. The total carbon is 3.3 to 3.4 per cent, the combined carbon about 0.75 per cent and the graphitic carbon is approximately 2.75 per cent. Chromium is a little less than 0.5 per cent.

While uniformity of analysis is always desirable, positive control of the rate of cooling is even more important. The centrifugal force of casting tends to produce a finer grain structure than sand casting would give, so the pearlite is finer than would be obtained had this iron been poured in the sand. Graphitic carbon is necessary, not for purposes of lubrication, but to prevent the ring from shrinking away from the steel stamping during cooling, as the solidification of a large amount of graphitic carbon is accompanied by an increase in bulk. Fig. 24, which is a photomicrograph of the same centrifugally cast iron unetched, at 100 diameters, gives a better idea of the uniform distribution of the small graphite flakes and reveals adequate strength in the iron, particularly as we know that it is supported by the steel shell to which it is fused.

Recent tests have developed some interesting facts as to the comparative stability of this iron and 2.5-per-cent-nickel iron. Some of us had thought that this pro-

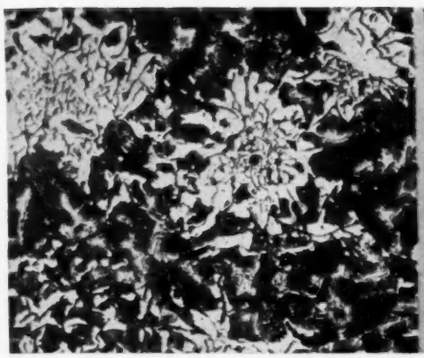


FIG. 22—UNDESIRABLE STRUCTURE OF CENTRIFUGALLY CAST BRAKE-DRUM
Magnification 100 Diameters

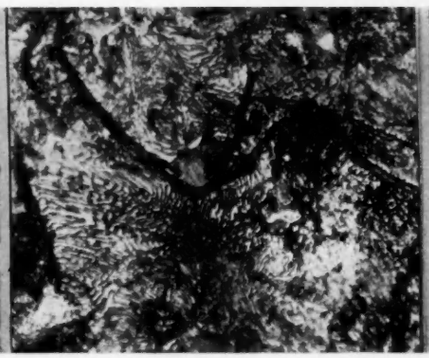


Fig. 23—Etched, Magnification 500 Diameters

Fig. 24—Unetched, Magnification 100 Diameters

DESIRABLE STRUCTURE OF CENTRIFUGALLY CAST IRON

portion of nickel is necessary to obtain thermal stability in iron so that the structure would not change at braking temperatures. Photomicrographs at 1000 diameters of both kinds of iron were taken. The samples were then subjected to 1200 deg. fahr. for 5 hr. and photographs were again taken. Very little change is to be seen in the grain structure of the plain iron, and tests on the brake-scoring machine certainly indicated that the change was not sufficient to reduce its resistance to scoring. The nickel iron is slightly more stable, and either should be satisfactory for use in centrifugally cast drums.

Only a portion of the difficulties with brakes can be attributed to the drums; and this paper has been limited to a discussion of scoring, which is only one of the problems connected with brake-drums. Heat dissipation, distortion and chatter are primarily questions of design, because they are controlled in turn by mass, shape and method of assembly; but abrasion, and more particularly that type of uneven wear known as scoring, is largely a problem in metallurgy.

No matter what types of drum and lining best meet the needs of a given chassis, a ferrous braking surface can be secured that has a grain structure suitable for its requirements. Under present conditions of brake operation, a drum surface composed principally of laminated pearlite should prevent difficulties from scoring. Such surfaces are seen in Figs. 8, 14 and 23.

THE DISCUSSION

A. S. VAN HALTEREN[2]—When the problem of producing a more efficient drum presented itself, the natural tendency was to improve the pressed-steel drum by making it from materials other than were commonly used at the time. Considerable success, some of it unexpected, was had in cold-stamping many of the newly tried materials. Some of them gave improved braking performance, but as a rule their high cost made them uncommercial.

Many materials were tried experimentally in our program. Drums made of certain materials would operate with less wear and less scoring than drums of low-carbon steel, but no change in fading or washout, bell-mouthing, expansion from heating or deformation under load was secured simply by changing the material of a stamped drum.

The next step was to investigate the possibilities of composite steel drums, in which the distribution of metal can be more advantageous because it is not limited by stamping practice. We produced braking rings of very good characteristics from various steels in various section forms which, when attached to suitable backs, made satisfactory drums.

One of the observations we made during these investigations was that the wear of the drum varies with differences in the microscopic structure of the steel. We have had excellent service from composite drums made from S.A.E. 5140 steel, and we believe that a correctly designed drum made from this material will perform satisfactorily on any passenger-car.

Full cast-iron drums have presented some very interesting metallurgical developments and have given a generally satisfactory account of themselves on large trucks and on motorcoaches. Of decidedly more current interest are the composite drums in which cast iron forms the braking surface. This type of drum affords the widest latitude in the choice of design and material. At least a dozen cast irons have been used experimentally.

Data have been collected from dynamometer tests on approximately 200 drums made by ourselves and other manufacturers. A typical test consists of 500 brake applications with normal load and 500 with 50-per-cent overload, all applications being of 8-sec. duration from a speed corresponding to 60 m.p.h. The relation of wear to load intensity, as determined from these tests, is charted in Fig. 25, which shows that the load intensity has a marked effect on the wear.

Fig. 26, the curves of which were obtained from dynamometer tests that provided no effect of road abrasives, shows that the hardness of the drum has but slight effect on wear.

The findings of our tests led us to make further investigations into the causes of wear. Tests with centrifuse and other drums had shown us that, irrespective of chemical analysis, a highly desirable feature of a good brake-drum was full pearlitic microstructure. From a close study of the structure in its relation to wear, we find that a drum lacking this full pearlitic structure does not give good wear and is likely to wear in waves or grooves. With the full pearlitic structure, a good wear curve is obtained and the braking surface remains flat and smooth.

Fig. 27 shows the kinetic energy per wheel of a 3000-lb. and a 4000-lb. car at various speeds. Considerable heat is generated at the juncture of drum and lining

[2] M.S.A.E.—Executive engineer, Motor Wheel Corp., Lansing, Mich.

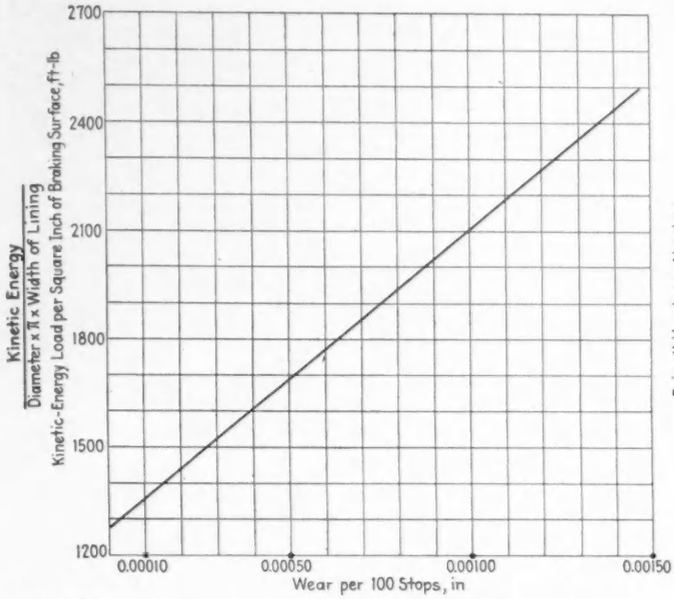


FIG. 25—RELATION BETWEEN LOAD INTENSITY AND WEAR

The Area Considered Is the Width of the Lining Times the Circumference of the Drum

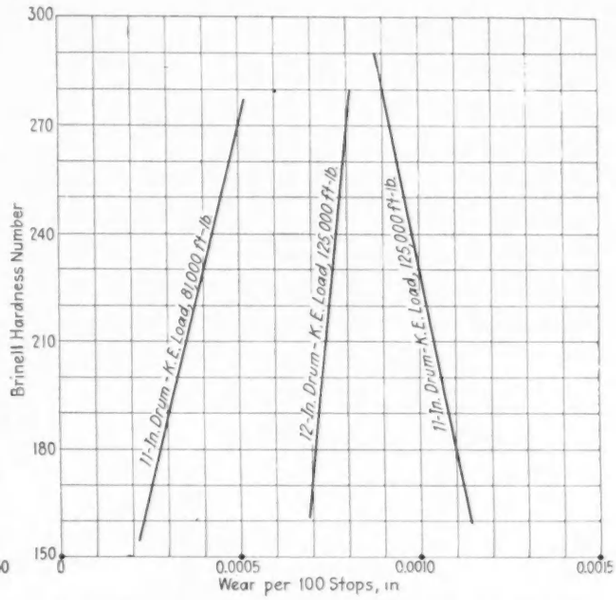


FIG. 26—RELATION BETWEEN HARDNESS AND WEAR

The Line at the Left Is for 81,000 Ft-Lb. of Kinetic Energy on an 11-In. Drum, the Center Line Is for 125,000 on a 12-In. Drum and the Right-Hand Line Is for 125,000 on an 11-In. Drum. Hardness Alone Seems To Have Little Effect on Wear

during a stop; and the resulting temperature rise of steel and cast iron, assuming all of the heat to be absorbed without loss by 1 cu. in. of the material, can be read by the aid of the respective straight-line curves. This chart also shows the weight of a brake ring to obtain a 100-deg.-fahr. rise in one stop. These weights correspond closely to good present practice in cast iron. Visualizing the high local temperaure led us to investigate the effect of heat on the drum, and this investigation soon centered on the problem of determining the

[3] M.S.A.E.—Chief engineer, Budd Wheel Co., Detroit.

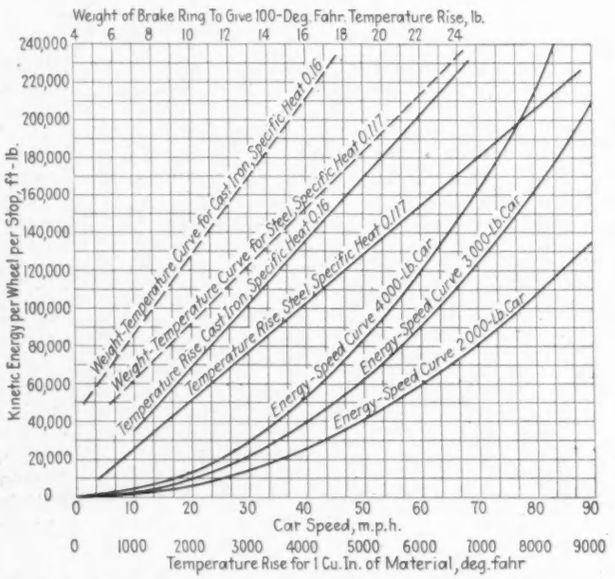

Weight of Brake Ring To Give 100-Deg. Fahr. Temperature Rise, lb.
4 6 8 10 12 14 16 18 20 22 24
Kinetic Energy per Wheel per Stop, ft-lb.
240,000
220,000
200,000
180,000
160,000
140,000
120,000
100,000
80,000
60,000
40,000
20,000
0
Weight-Temperature Curve for Cast Iron, Specific Heat 0.16
Weight-Temperature Curve for Steel, Specific Heat 0.117
Temperature Rise, Cast Iron, Specific Heat 0.16
Temperature Rise, Steel, Specific Heat 0.117
Energy-Speed Curve 4,000-Lb. Car
Energy-Speed Curve 3,000-Lb. Car
Energy-Speed Curve 2,000-Lb. Car
0 10 20 30 40 50 60 70 80 90
Car Speed, m.p.h.
0 1000 2000 3000 4000 5000 6000 7000 8000 9000
Temperature Rise for 1 Cu. In. of Material, deg. fahr

FIG. 27—RELATION OF KINETIC ENERGY TO CAR WEIGHT AND SPEED, TO WEIGHT OF DRUM TO GIVEN TEMPERATURE RISE AND TO TEMPERATURE RISE IN THE BRAKE-DRUM PER UNIT VOLUME

stability of the pearlitic structure under the application of heat. The results of our tests are illustrated by photomicrographs.

Fig. 28 shows the structure, as cast, of what we regard as a very good heat-resisting iron. In Figs. 29 and 30 is shown the same iron after heating for 2 and 12 hr. respectively at 1200 deg. fahr. The 2-hr. exposure has just begun to damage the structure, which is almost completely broken down after 12 hr.

The different behavior of another iron is illustrated by Figs. 31, 32 and 33, which show this iron respectively as cast, after heating for 1 hr., and after heating for 2 hr. at 1200 deg. fahr. It is obvious that this iron exhibits poor heat resistance. These two groups of photomicrographs represent the extremes.

Our experience is that a broken-down iron structure will crack in brake-drum service. Full cast-iron drums are known to crack in severe motorcoach service, and there is reason to believe that the heat effect indicated in Fig. 27 is the cause.

Our conclusions are that brake-drums should be metallurgically correct to resist wear, grooving, mat surfacing, excessive surface polishing and other forms of deterioration in service; the material should have correct microscopic structure and maintain that structure under severe heat conditions, stability of structure being a very necessary quality; and the material should be capable of commercial fabrication.

Sand Casting Reduces Cementite Segregation

C. L. EKSERGIAN[3]:—When Mr. Main states that the centrifugal force of casting tends to produce a finer grain structure than sand casting would give, I assume that he refers primarily to the grain structure of the free graphite and not of the pearlitic content. The grain size of the latter, within normal limits, seems to have little effect on wear as compared with that of the free graphite.

There seems to be little reason to believe that grain size is a function of internal pressure during cooling. This observation is supported by a comparison between

centrifugal and permanent-mold casting, which may be regarded as similar if we omit the elements of internal pressure and turbulence that are peculiar to the centrifugal process. We find that we can obtain similar structures by these two methods, which indicates that grain refinement is not a function of internal pressure.

Everything else being the same, the centrifugal casting will tend to exhibit a more refined grain structure than the sand casting. This is attributable to the fact that the cooling rate with the centrifugal process is generally higher than with sand casting, rather than to internal pressure. However, the same degree of grain refinement can readily be obtained in the sand-casting process by slightly varying the carbon-silicon ratio. Certain inherent tendencies in this method of casting react favorably in obtaining the desired structure.

Free cementite has considerable effect on wear. If it is too great and is not well dispersed, being segregated in relatively large lumps, hard spots will result, with machining and scoring difficulties following. The tendency toward this type of segregation is less pronounced with the lower cooling rate of sand casting. The strain gradient is less with the lower rate of cooling. Thus a more stable state resulting in less deformation should be expected in a sand casting.

Unquestionably the most conspicuous natural advantage of sand casting lies in its easier controllability for obtaining commercial consistency. Uniformity between one piece and the next is of utmost concern with all castings, and particularly so in brake-drums. Because of the natural variations, such as in pouring temperature, which attend any casting process, the lower the cooling rate is, the less likely are these normal variations to be reflected in the final structure.

My remarks have been on limiting tendencies, rather than on final limitations. If the limiting tendencies are obstacles, the difficulties of overcoming them are enhanced and the chances of obtaining consistency are reduced.

Assuming that the centrifugal process as now practiced is representative of what may be expected, our experience indicates that a great uniformity exists in sections of the same drum and in successive drums when sand casting is employed.

Temperature Recovery Is Vital

I agree with Mr. Van Halteren that mass must be taken into account to control temperature rise. However, mass is only one of several factors. The functions of a brake-drum are to convert work in the form of mechanical energy into work in the form of heat and subsequently to dissipate this heat as rapidly as possible. While increments of heat are entering the drum, decrements are leaving simultanously by way of conduction, radiation and convection.

The instantaneous temperature of the drum is a function of both the absorption factor and the heat-dissipating properties of the drum. If heat could be removed as fast as it enters, no change in temperature would result. As this balance cannot be achieved, a condenser effect is introduced in the form of mass, which serves to absorb part of the heat input. If the design of the drum is such as to raise the rate of dissipation, the temperature rise per given mass of drum will be reduced.

Our experience indicates that, holding both the temperature rise and the kinetic energy constant, the required mass can be appreciably reduced by improving the heat-dissipation factor in the design. The employ-

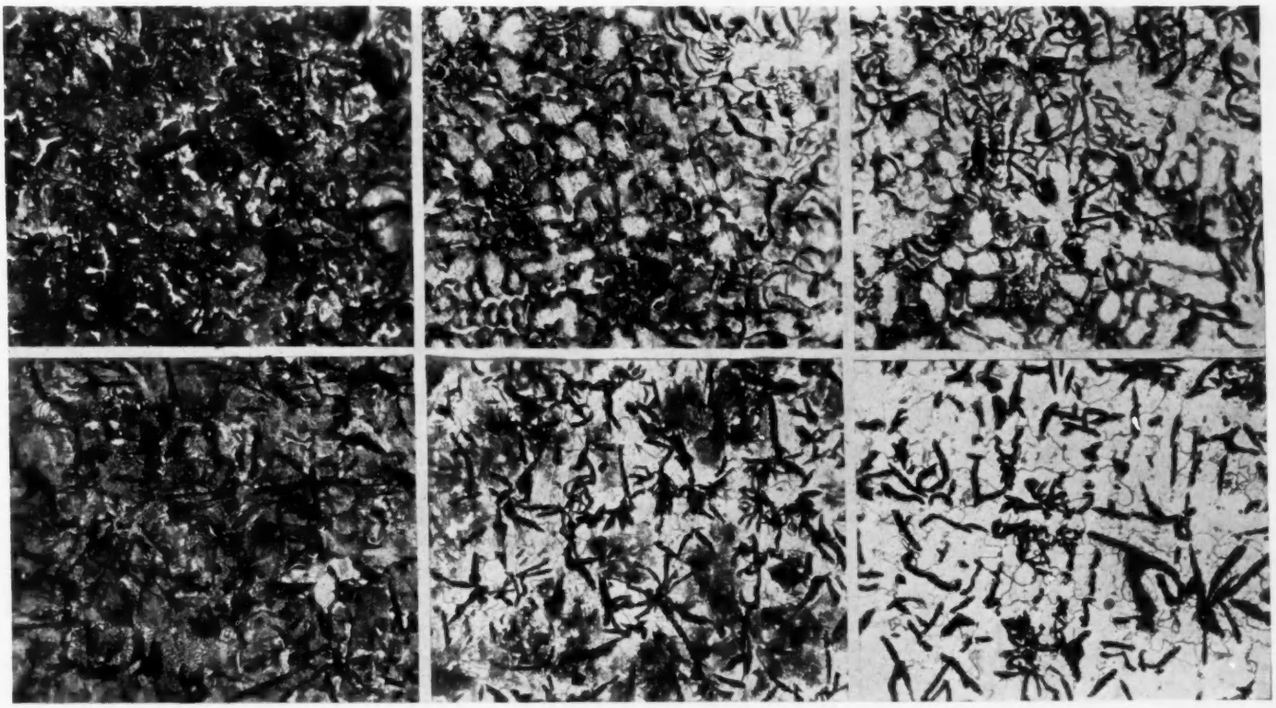


Fig. 28—The Iron as Cast

Fig. 29—After Heating for 2 Hr. at 1200 Deg. Fahr.

Fig. 30—After Heating for 12 Hr. at 1200 Deg. Fahr.

(Above) Micrographs of an Iron That Resists Heat Well—Magnification 100 Diameters

Fig. 31—The Iron as Cast

Fig. 32—After Heating for 1 Hr. at 1200 Deg. Fahr.

Fig. 33—After Heating for 2 Hr. at 1200 Deg. Fahr.

(Below) Micrographs of an Iron That Resists Heat Poorly—Magnification 100 Diameters

ment of a higher dissipation factor, instead of increasing the mass, results in a distinct advantage in realizing better cooling recovery, in addition to the savings in unsprung weight and costs. Under severe driving conditions, braking is more likely to become critical because of its intensity and frequency of recurrence rather than because of prolonged application. Obviously, the ability to recover is of utmost concern in this type of driving, if not in most driving. The employment of greater mass penalizes the cooling recovery.

D. P. FORBES[4]:—The ideal structure in cast brake-drums is pearlitic or sorbitic. Any trace of free ferrite tends toward rapid wear or scoring. On the other hand, an excess of free cementite results in rapid wear of the linings, since the softer pearlite is worn away, leaving the hard cementite in relief to act as a cutting-tool. Graphite in cast drums acts only slightly as a lubricant; its chief function is to break up the continuity of the structure, thus preventing metallic flow of the surface and the tendency to score. However, a structure containing patches or areas of extremely fine graphite in a sort of lacework design is to be avoided, since this usually contains quantities of free ferrite. We have never found that the size of the graphite flakes made much difference provided they are uniformly distributed in a matrix of suitable structure.

Alloys are of advantage chiefly when they contribute to the strength of the metal, since wear resistance seems to depend more on strength than on hardness. Alloys can be omitted if suitable material can be obtained by metallurgical control.

Fading in brakes depends more upon linings than upon drums, although steel drums under some conditions seem to be less subject to this action. Possibly this is because a small amount of work is done by scoring or metal flow on the surface of the steel during a stop from high speed. Possibly a lining could be developed having a coefficient of friction that would rise with higher temperatures.

Our observations have shown that a Brinell hardness slightly over 200 seems to give the best all-round properties. Greater hardness seems to cause more rapid lining wear.

The heating of brake-drum metals to elevated temperatures in a laboratory will naturally cause a breakdown of the combined carbon, with a loss in wearing properties; but drums that have been badly overheated in service, with resultant heat-checking and discoloration, do not show any perceptible reduction of combined carbon.

Recently we have turned our attention to the development of a brake-drum metal having strength and ductility suitable for integrally cast hubs and drums. We have succeeded in producing a metal having a tensile strength of 100,000 and a yield point of 65,000 lb. per sq. in. and 6 to 10-per-cent elongation. This metal contains about 1½ per cent of free graphite and has a sorbitic matrix.

Tests so far indicate a reduction of 20 to 25 per cent in weight of the combined hub and drum unit as compared with present practices, with greater strength and more perfect maintenance of concentricity.

[4] M.S.A.E.—President, Gunite Foundries Corp., Rockford, Ill.

Centrifugal Casting Gives Greater Uniformity

F. L. MAIN:—We agree with Mr. Eksergian that, together with temperature control, the carbon-silicon ratio must be varied to obtain the same kind of grain structure in sand casting as in centrifugal casting, but we find that in sand casting the lamellar pearlite is coarser than in centrifugal casting. We have obtained a more uniform product by centrifugal casting, for, as between a number of drums of the same design, we obtain uniform volume, absence of shrinks and pockets; and foreign matter, being lighter than the iron, is thrown to the inside and is removed when the braking surface is machined.

Our experience bears out Mr. Forbes's contention that the graphite flakes should be of small size and uniformly distributed, but to obtain this result in centrifugal casting we have not found it necessary to alloy; the analysis giving the correct pearlitic structure also happens to provide a satisfactory graphite distribution. Tests have indicated, however, that the use of manganese in excess of 0.84 per cent in steel rings and of molybdenum in cast drums improve the wear factor. We have found that the best results are obtained when the tensile strength is from 40,000 to 45,000 lb. per sq. in., and the Brinell hardness from 195 to 210.

W. S. James has suggested that tests of drum materials at advanced temperatures may develop significant data as to stability, strength and wear. We are now conducting tests to determine the effect of sustained braking temperature on drum life and performance.

Corrections in Boerlage and Broeze Paper

TWO corrections should be made in the paper entitled Ignition Quality of Diesel-Fuels as Expressed in Cetene Numbers, by G. D. Boerlage and J. J. Broeze, which was preprinted in advance of the Semi-Annual Meeting and published in the July, 1932, issue of the S.A.E. JOURNAL. Instead of using mesitylene as stated at the top of the second column of p. 286 of the JOURNAL (p. 4 of the preprint) alpha methyl naphthalene is now being used. In the seventh line under the cross head Nuclear Ignition in the second column of p. 290 of the JOURNAL (p. 8 of the preprint) the reference should be to "process II" instead of to "process I".

Free-Wheeling Corrections

THE caption of Fig. 11 in Austin M. Wolf's paper, which was printed in the July issue of the S.A.E. JOURNAL, p. 270, should say that the free-wheeling unit in the Oldsmobile transmission is a Detroit, as stated in the text and in Table 2, instead of a Warner.

Credit for the drawing of the Millam free-wheeling device reproduced as Fig. 22 in the paper should have been given to *The Motor*, of London.

Conclusions from Headlight Research[1]

Report by H. C. Dickinson[2] and H. H. Allen[3] on Bureau of Standards Series of Tests

A RATHER extended research project was undertaken at the Bureau of Standards in 1927 and 1928 at the request of the National Automobile Chamber of Commerce and supported largely by funds appropriated by that organization. The program consisted entirely of physical measurements of one sort and another, all made on highways under various conditions by a limited number of observers. When the results were analyzed, the conclusions to which they led proved to be controversial in various respects and consequently publication of the results has been long delayed, except for a short paper[4] published in January, 1929. The brief discussion of the methods and results contained in the present paper has been prepared with a view to including only such results and conclusions as are based on measurements. This, of course, does not apply to suggestions for further research.

Tests were of four kinds, explained here very briefly, although the observations, running into many thousands of separate determinations, involved two years of actual time.

Visibility Distances on Straight, Smooth, Level, Concrete Roads

The object of the tests for visibility distances was to determine the actual distance at which typical objects could be seen on a straight, level, concrete road with head-lamp beams of varying vertical and horizontal spread, vertical tilt and bulb candlepower, all without opposing lights. The equipment used for this purpose is illustrated in Figs. 1 and 2, Fig. 1 showing at the right a test car on which were mounted four separate lamps capable of adjustment in all the respects noted, and Fig. 2 showing two of the test objects employed. The beam patterns employed were all symmetrical in horizontal distribution. Several thousands of observations were made under the widest practicable range of conditions, and the following conclusion were reached:

(1) The distance at which objects can be seen is greatest when the maximum intensity is aimed horizontally, drops off sharply when the beam is depressed below this position and somewhat less sharply when the beam is elevated.

(2) The visibility distance decreases with increased spread of the beam either vertically or horizontally if the bulb candlepower remains constant, because the light intensity on the object decreases.

(3) The visibility distance decreases with decreased intensity, whether because of greater spread of beam, lower candlepower or poorer reflection, by an amount roughly proportional to the cube root of the intensity of light falling on the object.

Thus, if an object could be seen at a distance of 400 ft. with a given beam intensity, a visibility distance of 500 ft. could be secured only by using double the candlepower in the bulbs or reducing the beam 50 per cent in area. (Some unpublished results obtained elsewhere show even less change in visibility distance for a given change in candlepower.)

(4) The visibility of different objects varies with their size and more particularly with the color and character of the surface of the object.

In general, for the same lighting conditions, a white object was visible for twice the distance for the same object when black. This difference is less than would be expected and probably is due partly to the background conditions which obtained in these tests. (More accurate figures for these effects have since been obtained by other observers, but these results do not affect the general conclusions.)

(5) Visibility distance does not depend to any significant extent upon the speed at which the object is approached.

(6) A comparison between results obtained with and without moonlight showed no significant differences in visibility distance under these two conditions.

The fact should be kept in mind that the foregoing conclusions apply directly only to the limited conditions of a straight, smooth, level, concrete road. Some of them are greatly modified when applied to results obtained under a variety of road conditions, as explained below.

Visibility Distance in General, Under Average Hard-Surfaced Road Conditions

For the purpose of extending the observations to cover average road conditions, both with and without opposing lights, special equipment was devised and used each night for several months to collect the great

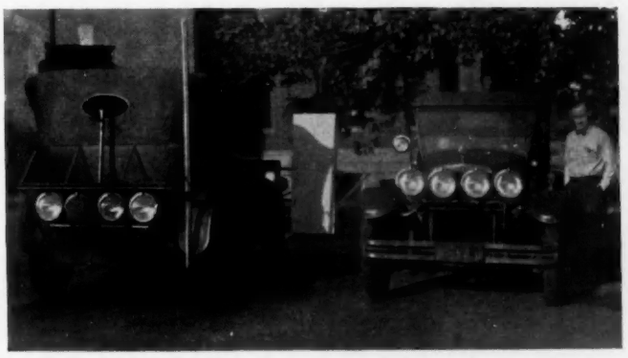


FIG. 1—TEST CARS USED IN HEADLIGHT RESEARCH

Car *A*, at Right, Was Fitted with Four Separate Head-Lamps Capable of Adjustment for Varying Vertical and Horizontal Spread of Beam, Vertical Tilt and Filament Candlepower. It also Carried a Stadiometer for Quick Measurement of Visibility Distances. Car *B*, at Left, Had Exactly Similar Equipment Mounted on the Rear and Provided with a Black Curtain That Could Be Lowered To Make the Light Visible. It also Carried on a Right-Hand Door a Special Test Object That Could Be Swing Out into View, as Seen between the Cars, and Signal Lamps and Markers To Aid in Observation

[1] Approved for publication by the Director of the Bureau of Standards, Department of Commerce.

[2] M.S.A.E.—Chief, heat and power division, Bureau of Standards, City of Washington.

[3] Assistant mechanical engineer, Bureau of Standards, City of Washington.

[4] See *Transactions of the Illuminating Engineering Society*, vol. 24, no. 2, p. 15.

mass of statistical data on which the following conclusions are based.

The equipment consisted of two cars, *A* and *B*, shown in Fig. 1. Car *A* was fitted with the special adjustable head-lamp equipment used in the previous tests, and Car *B* with an exactly similar equipment mounted facing the rear and provided with a black curtain that could be lowered over the head-lamps to render them invisible when so required. This car also carried a special test object mounted on a right-hand door so that it could be swung out into view, as shown at the center in Fig. 1, when observations were to be made. Car *B* also carried certain signal lamps and markers to assist in observation. Car *A* carried a stadiometer, whereby distances could be measured quickly at all times. In this entire series of tests the head-lamps on the two cars were always adjusted to the same vertical and horizontal beam spread and to the same tilt as referred to the axis of the car.

The test procedure was to drive Car *B* to some chosen location on the road, draw to the left and stop, with all lights out except a safety red light on the front, and the test target swung out so as to be near the center of the highway.

Car *A*, with its driver and observer, then approached, with head-lamp switched on and adjusted for some one of the many conditions to be tried, until the test object became just clearly distinguishable. Then, at a signal, a small target lamp on Car *B* was switched on and a stadiometer reading taken. Next, the rear head-lamps on Car *B* were uncovered and turned on and Car *A* approached nearer until the test object again became distinguishable in the glare of the opposing lights. Again the distance was measured and recorded and Car *B* proceeded to a new location. Throughout a large part of the program, measurements were also made of the illumination actually falling in the eyes of the approaching driver at this point. This was done with a portable photometer mounted on the windshield of Car *A*. Observations had to be made on unfrequented roads or late at night to avoid interference with other traffic and interference of other traffic with the observations.

The results of the long series of observations made with this equipment include the widest practicable range of road conditions and a wide range of weather conditions, all of which are given something like their correct weight. Therefore these observations represent a fair approach to the average conditions met with on the road, except that there was only one opposing car and that was stationary.

The conclusions drawn from the tests may be stated as follows:

FIG. 2—OBJECTS USED FOR TESTING VISIBILITY

(1) Horizontal spread of the beam, using the same bulb candlepower, has a small effect on the average visibility distance, not more than 20 per cent for spreads ranging from 8 to 40 deg., probably owing to change in beam intensity with spread.

(2) Brightness of beam has much less effect on average visibility distance for all sorts of road conditions than for straight roads. For an average visibility distance of 300 ft., more than a five-fold increase in the candlepower would be required to increase the distance to 400 ft. Thus, if 300 ft. corresponds to 21-cp. bulbs, more than 100 cp. would be needed to give an average visibility distance of 400 ft.

(3) The average visibility distance under all conditions of road and horizontal beam width is a distinct maximum when the maximum intensity of the beam is aimed horizontally, and drops to 50 per cent for a 4-deg. depression and to 70 per cent for a 4-deg. elevation. This decrease in visibility distance is less for beams having greater vertical spread.

Effect of Opposing Lights

Results with opposing lights, as is to be expected, differ radically from those without such disturbing factors. The results obtained indicate the following conclusions:

(1) Increase in spread of the beam from 8 deg. to 40 deg. reduces the visibility distance on straightaways and curves to the right by 15 to 20 per cent. For curves to the left the effect is negligible.

(2) Vertical tilt of the lamps, altogether contrary to the usual opinion, has no significant effect on the visibility distance of objects on the highway a short distance beyond opposing lights.

The apparent discrepancy between these measurements, which are not open to question, and the general opinion, which doubtless is based on experience that seeing is better with lamp beams depressed, may be found in the assumption that in passing other vehicles the driver is most concerned with visibility of the road shoulder to the right and not of possible obstructions in the traffic lane. Road-shoulder visibility is improved by lowering the beam. In this respect the driver benefits more by depressing his own head-lamp beam than by depression of the opposing beam.

(3) The actual distance at which objects in the right of way just to the rear of approaching headlights can be distinguished is surprisingly constant for all conditions encountered. This distance seldom exceeded 120 ft., and seldom was less than 80 ft. The average of all observations was about 100 ft., and few of them departed significantly from this figure. The fact should be noted, however, that all observations were made with similar head-lamps similarly aimed and adjusted on the two vehicles.

(4) Within the limits just stated, the visibility increases somewhat with decreased amount of light striking the driver's eyes, as measured by the portable photometer.

When it is remembered that the eyes of the driver are at a level not much removed from that of the test object designed to represent another vehicle or a person on the highway, it is evident that elevating or depressing both sets of beams had the effect mainly of changing the general light level, which it is known does not greatly affect visibility.

(5) The characteristic differences between the results on curves to the right and curves to the left are revealed in the observations with opposing

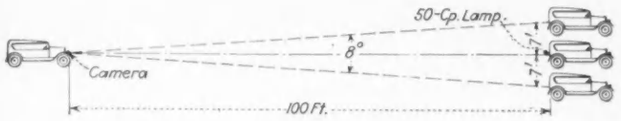


FIG. 3—DIAGRAM OF METHOD OF OBTAINING RECORDS OF VERTICAL DISPLACEMENT

A Rear Lamp on the Car Ahead Traced Lines on Photographic Plates in a Camera on the Following Car

lights, but, as noted above, all observations lie within the narrow range of 80 to 120 ft. visibility distance.

Necessary Vertical and Horizontal Beam Spread

Since the light intensity and therefore the visibility distance are affected by the horizontal and vertical spread of the head-lamp beam for a given bulb candlepower, it is important to know what should be the minimum vertical and horizontal spread of a satisfactory head-lamp beam. To secure data on the subject, Cars *A* and *B* were used as follows:

On Car *A*, an ordinary camera was mounted rigidly on the head-lamp mounting and fitted with a shutter that could be operated from the driver's seat. Car *B* was fitted with a single low-power lamp mounted on the rear near the center of the body. Thus, when Car *B* was driven ahead of Car *A* at any selected distance, with the camera shutter open, a fine line was traced on the photographic plate, giving the angular relationship of Car *B* to the aim of Car *A*, which was following it. If the road turned to the right, the trace travelled to the right, and vice versa; while for hills and valleys the line rose and fell. Moreover, spring action or road irregularities affecting Car *A* were recorded because of the changes in axial direction, as illustrated in Fig. 3.

By this means records were obtained showing the various angular positions that would have been required to keep a narrow spotlight beam constantly on the center of the car ahead. It is clear that the vertical and horizontal angular spreads obtained represent directly the angles that must be covered by a head-lamp beam to directly and continuously illuminate the vehicle ahead on the particular stretch of road on which the photographs reproduced in Fig. 4 were taken.

A large number of such plates were made with the two cars held at distances of 100, 200, 300 and 400 ft. from each other at speeds varying from 20 to 40 m.p.h. and on a wide variety of roads. For each of the four distances a plot was made giving the vertical and horizontal scale of each print in angular degrees, with a figure of the distant vehicle also in its correct angular dimensions. Two of these plates are shown in Fig. 4. The bottom record is from an exceptionally smooth, flat road.

A study of these figures shows that a vertical spread of at least 6 deg. and a horizontal spread of at least 30 deg. is necessary for the head-lamp beam to fall on any part of a car 200 ft. ahead for 90 per cent of the time.

The most striking conclusion drawn from these results, however, is that the very sharp cut-off at the top of the beam and the accompanying emphasis on exact aiming, which were in vogue a few years ago, had no rational basis in headlight performance. Even on the smoothest level road the beam is continually sweeping through an arc of 3 or 4 deg., which at a distance of 200 ft. includes nearly twice the vertical height of the car. Obviously, a sharp cut-off of the beam cannot be confined to the space of some 4 ft. between the level of the road and the eyes of the driver at distances of 200 ft. or more. This factor assumes greater importance when one considers that visibility distance is proportional to the size of the object and that the size of that portion of such objects as persons, animals or horse-drawn vehicles which is visible within 1 or 2 ft. of the road surface is very small indeed.

The results of this series of tests were published in January, 1929, in a paper entitled, Automotive Headlight Requirements from the Driver's Point of View[5]. This paper also included an analysis of the horizontal beam-spread necessary for visibility of the road on curves and so on, as related to car speed, showing that for high speeds a spread of 20 to 25 deg. will cover all the field of view that is unobstructed by curves and the like, while at the low speeds necessary on very sharp turns a beam spread of 100 deg. may be needed.

Glare and Roadside Illumination

A special series of experiments was performed to obtain a relationship between intensity of illumination in the driver's eyes and that necessary to be thrown on the road shoulder so that it could be seen.

Two cars were placed on a concrete roadway a fixed distance apart, as shown in Fig. 5, the cars remaining stationary. The dirt shoulder was flush with the concrete at a fixed point. A fixed intensity of illumination was focused directly into the observer's eyes. If the intensity of illumination on the road shoulder was not sufficient to render the shoulder clearly visible at the instant the light came into the observer's eyes, it was increased until the shoulder became visible. The light entering the observer's eyes was then turned off, the spot on the road remaining, and the process was repeated after an interval of 15 to 20 sec. When the series had been completed, the intensity of illumination from the observer's car that fell on the road shoulder was measured. Several repeat tests were made of each set of observations to obtain check results.

Similar observations were made of the spot of road shoulder illuminated, increasing by 5-ft. intervals the

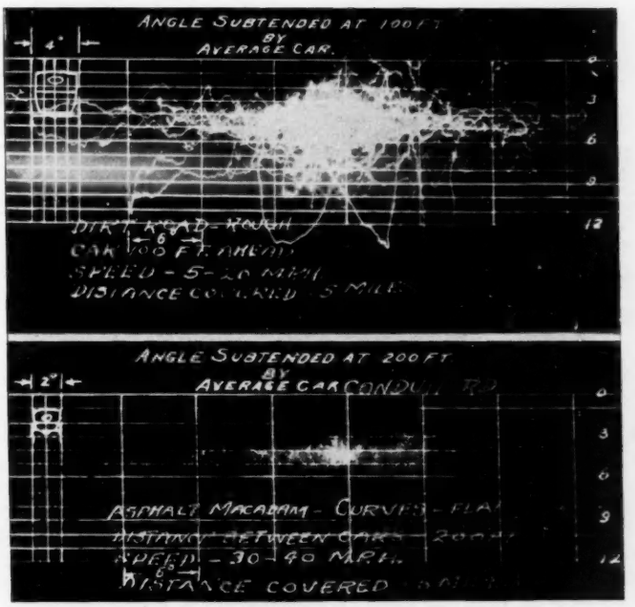


FIG. 4—TYPICAL PHOTOGRAPHIC RECORDS SHOWING VERTICAL AND HORIZONTAL ANGULAR SPREADS REQUIRED

The Scale of Each Print Is Given in Angular Degrees in the Plot at the Left, Together with a Figure of the Leading Car in Correct Angular Dimensions

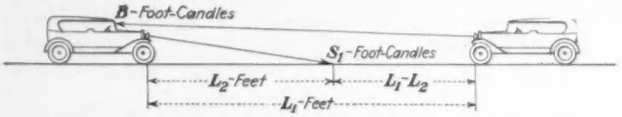


FIG. 5—DIAGRAM OF SET-UP USED IN GLARE TESTS

The Relationship between Intensity of Illumination in the Driver's Eyes and That Necessary To Render the Road Shoulder Visible Was Found by Varying the Intensity of the Light on the Road Shoulder

distance from the observer up to the total distance between cars, which remained fixed throughout. Also an exactly similar series of tests was made, each series with an increased intensity of illumination in the driver's eyes. One observer made all the observations.

Data from other sources indicate that the intensity of illumination on the road shoulder for a given intensity in the driver's eyes varies widely, although the probable distribution may be calculated from these data.

The conclusions based on these experiments are that:

(1) For distances within 25 ft. of the driver, the intensity of illumination, in foot-candles, falling on the road shoulder must be 1¼ times as great as that falling in his eyes, so that he can see the road shoulder immediately after the light comes into his eyes. For distances up to 45 ft. this ratio is about 5; and for greater distances the ratio is increased enormously.

(2) If, instead of requiring that the driver see the road shoulder immediately after the light comes into his eyes, a time lag is allowed, a less intensity of illumination falling on the road shoulder is required, and the greater this time lag is, within limits, the less the road shoulder intensity will have to be. The effect of this time lag will be greatly different for different persons, as well as greatly different for the same person from one time to another.

(3) If the intensity of illumination at the driver's eyes be increased beyond a given amount, a point is reached where no amount of increase in road-shoulder illumination will suffice to give instantaneous visibility thereof.

For the observer conducting the tests reported, and for the condition of his eyes reported at that time, this limit was somewhat above 4 ft.-candles, and all the tests conducted were well below this intensity. The brightest head-lamps that one is likely to meet on the road could not, in all probability, reach this limit at distances greater than 100 ft. between cars. For this or shorter distances the chance of meeting the full intensity of approaching lights is small.

General Conclusions Deduced from Tests

The more significant general conclusions drawn from the tests are as follows:

(1) For the best conditions of visibility for open-road driving at good speeds in the absence of opposing lights, head-lamps should be aimed with the maximum intensity horizontal and should have vertical and horizontal beam-spreads of about 6 and 25 deg. respectively.

(2) For meeting other vehicles and for slow speeds on rough or very crooked roads, foreground illumination within the first 100 ft. should be increased and a much greater horizontal beam-spread up to about 100 deg. is desirable, to give good roadside illumination and lighting around curves. In these cases the top of the beam should be depressed below the horizontal.

(3) No practicable sort of symmetrical distribution will enable both approaching drivers to distinguish objects to the rear of the approaching lights for distances much in excess of 100 ft. This emphasizes the fact that, in meeting other cars, the driver depends for assurance of a clear road upon the fact that the opposing car has, within the past few seconds, lighted the road over which he is about to pass. Passing a car at rest involves much greater danger than passing one in motion.

(4) The most important element in avoiding danger in meeting other cars is adequate light on the roadside. Since ability to see to the rear of the opposing car is almost negligible, curb illumination must suffice and therefore must be sufficient for safe visibility when facing the unavoidable glare.

(5) By far the most serious problem involved in safe headlighting is the great disparity in brightness between different lamps. Beam intensities vary more than 10:1 even in fairly normal equipment. While this difference accounts for less than a 2:1 difference in visibility distance without opposing lights, it results in almost complete lack of visibility for the driver using dim light when meeting bright lights, no matter what measures are taken to reduce glare. Nine-tenths of the glare problem probably would be solved if the brightness of lights could be kept within a factor of 2 or 3:1.

Future Problems

So far as headlighting is concerned, it is important to unify the results that have been obtained and, if they are found to be correct, to apply them to revision of head-lamp equipment and of legislation, so as to permit the use of equipment that is better adapted to safe seeing.

Means should be found for reducing the present great disparity in brightness between different head-lamps. If it were practical to limit this range of brightness to a ratio of 3:1 in. beam candlepower, particularly by eliminating dim lights, both old and new, a very long step could be made toward solving the headlight problem.

One of the most immediate practical needs, however, seems to be a study of the effect of various types of artificial highway-lighting on safe visibility. Almost unquestionably certain conditions exist under which some types of fixed highway-illumination introduce real hazards as compared with headlighting alone, even in the present state of headlighting inefficiency. For instance, in any field of fixed illumination by scattered units, there are necessarily some locations where an object will be illuminated in such a way as to offer no contrast to its background and hence will become practically invisible. These conditions should be studied and their importance determined.

The presence of any exposed light has the effect of greatly reducing the visibity of objects that are dimly illuminated by another source, such as head-lamps; therefore, in some cases, objects readily visible by headlighting alone become less visible when fixed lighting is applied.

THE DISCUSSION

W. C. BROWN[5]:—Dr. Dickinson makes no specific recommendation regarding distribution of light within the beam. Apparently his experiments were conducted with beams relatively uniform in candlepower laterally and

[5] M.S.A.E.—Illuminating engineer, Nela Park engineering department, General Electric Co., Cleveland.

with a vertical distribution having the highest intensity at the center of the beam. Modern practice recognizes the distinct advantage of having a vertical distribution that places the maximum intensity at or just below the horizontal, this high-candlepower part having a depth of about 2 deg. or less with present aiming restrictions. The intensity below this part drops off rapidly as the beam approaches the car. Likewise, a uniform lateral distribution is not most desirable with the limited amount of energy available. Concentration of a considerable portion of the light down the center of the road is desirable. In practice, this high-candlepower part is from about 5 to 10 deg. in width. In recent years a demand has arisen for more light farther out to the sides, and this has been met by the designers of equipment. In doing this, however, some intensity is sacrificed down the center of the road, hence the real need for 32-cp. lamps and their ready adoption by motorists everywhere.

Brightness of Head-Lamps in Use

Dr. Dickinson suggests that much of the glare problem would be solved if the brightness of head-lamps in service could be kept within ratios of 2 or 3:1. This seems impossible with the present current-regulated battery-generator systems. We have made a great number of measurements of voltage on cars in actual service and find that, for the large group of cars that are fairly close to the average, a variation in head-lamp brightness of as much as 3½:1, merely because of high and low voltage, is not uncommon. Considering the entire range of cars, this variation in brightness due to voltage alone would be as much as 7½:1. Added to this voltage variation are many other factors, such as the collection of dust and dirt, which make the reducing of the range of brightness a rather involved matter.

A series of tests is reported in the paper as taken with one observer. With such tests, one should be very sure that the observer had normal vision. Even then an individual is a poor "seeing meter" and, to obtain results which are at all conclusive, integrating a large number of observations is necessary, or better, to have a great number of observations taken with a considerable number of observers.

An Asymmetric Beam for Passing

Regarding the author's statement that

> no practicable sort of symmetrical distribution will make it possible for both approaching drivers to distinguish objects to the rear of the approaching lights for distances much in excess of 100 ft.

an asymmetric distribution can be provided when passing approaching vehicles, with more light on the right-hand side of the road, possibly up to the horizontal, with the highest intensity at or near the top. Our experiments indicate that such a beam should cut off fairly sharply along its left edge and that, for passing, the light directed to the left side of the road should be similar to that provided by the conventional depressed beams. While it is true that, when going around a right-hand curve, such a beam would strike the oncoming driver in the eyes, the period of time when he is in the full-intensity portion would usually be short and his line of vision would be directed toward the road shoulder on his right and at a considerable angle to the approaching beam, hence glare thus produced will not be greatly disturbing.

Such an asymmetric distribution, of a desirable character, can be obtained with specially designed auxiliary units, of which I understand at least two are now on the market; or it can be incorporated in the design of the head-lamps themselves, as has been done on one of the 1932 passenger-cars.

Good Highway Lighting Improves Visibility

Although some highway-lighting systems in use today afford no increase in visibility of objects on the road, or may even make seeing more difficult, I should say that in every case these are poorly designed systems in which modern engineering developments of lighting equipment and knowledge of their use have not been employed. Any highway-lighting system which is representative of good practice will very definitely improve the visibity under all conditions of road surface and weather, so much so as to be far above the threshold values of seeing employed by Dr. Dickinson in his experiments. Correctly designed highway lighting provides a silhouette effect of great importance in improving visibility of objects on the highway, although this effect is impossible to measure or express numerically.

The question today is not so much how to design and install a suitable highway-lighting system as the volume of night traffic that will economically justify the cost of the lighting. Under what might be called "average" conditions, good highway lighting costs a total of the order of $1,200 per mile per year. Such a cost is clearly justified on intercity routes carrying heavy night traffic, and on which the majority of night accidents occur, but the lower limit of traffic volume that justifies highway lighting still remains to be determined. A rather complete accident survey made by the State of Indiana indicates that on rural highways the accident hazard is relatively four times as great by night as by day. Such factors are important in weighing the value of adequate highway lighting.

The Illuminating Engineering Society has an active highway-lighting committee which has established the minimum lighting requirements for safety, and these have been published in code form. The committee is constantly endeavoring to improve both equipment and its application, and I am sure would welcome some cooperative activity with the S.A.E. Headlight Research Committee.

Should Educate Public with Motion Pictures

W. M. JOHNSON[6]:—The motion pictures shown by Dr. Dickinson offer one of the most practical means I have ever seen of explaining to the layman why head-lamp beams glare and why dual-beam head-lamps are necessary.

During my several years' experience in automotive lighting, I have talked with hundreds of persons regarding the problem of head-lamp beams bobbing up and down because of the action of springs and tires on the vehicle and the irregular contour of the roads, yet I am certain that none, with the exception of lighting engineers, has ever really understood it. Most persons adhere to the impression that glare is caused by some mysterious property of the lens, reflector or filament and can be cured only by some magic applied to these, or else is inherent in the name of the device or the number of patents issued to its designer.

Dr. Dickinson's method of using daylight motion pictures taken through an aperture representing a head-lamp beam illustrates the problem so clearly that I do not see how anyone could fail to understand it. I believe we have spent too much time in trying to educate the public in the technical details of lighting, whereas, with the competent designers of lighting equipment that we have today, we really need to teach the motoring public only two things:

(1) How to keep the lighting equipment in proper working order

[6] M.S.A.E.—Sales department, incandescent lamp department, General Electric Co., Cleveland.

(2) How and when the passing beams should be used when meeting other cars

I think that both of these points could be put across in a very short time with such a motion picture, if the picture includes only the pertinent points. A 1000-ft. reel of sound pictures should be adequate if copies were made widely available to all educational bodies such as schools, safety leagues, other civic bodies and State enforcement authorities. It could also be used to advantage by motor-car distributors, dealers and service stations. Such a film, in addition to showing clearly the need for using the lower beams to prevent glare and the upper beams for maximum safety when no approaching vehicles are in sight, would also show when to switch to the lower beam and when to switch back to the upper beam.

I believe that the potentialities of this method of education are enormous, and I suggest that the Society, through its Headlight Research Committee, sponsor the production of such an educational film. This, in my opinion, would go farther toward improving night driving conditions than anything heretofore attempted.

The Use of Trailers with Motor-Trucks

(*Concluded from p.* 322)

sure sales of trailers by trailer companies nor because some truck companies have recently gone into the manufacture of trailers. It is primarily because of the economics of the situation and incidentally because of legislation. Any economically sound method of doing anything is bound to come to the front in spite of railroads or legislation.

In saying that trailers are more commonly used incidentally because of legislation, I mean the economics of the law, not the law itself. For example, it is possible to build a large, single four-wheel vehicle that will carry more than many of our present tractor-trailer units; in fact, we have done it and still do it for those who operate on their own property, as in quarries and on large construction jobs as it is economically sound.

But, in building a National network of highways, we are informed by the Bureau of Public Roads that any highway which will withstand the elements must be made a certain thickness and that such a road will safely carry all of our automobiles and four-wheel motor-trucks up to about 3-tons pay-load capacity. Furthermore, by spending only 10 per cent more at the time of construction, the same road can be made strong enough to carry 5 tons. Going farther, to make that road strong enough to carry four times as much on four wheels would increase the cost of the highways beyond consideration today. So we find that, by distributing this increased load on additional axles and with proper spacing of those axles, we can economically build up both the roads and the motor-vehicles to transport these great loads.

The crux of the situation is that a thing which is economically sound will come to the front, but we as an industry cannot wait until that develops; we must get together and protect and even father that which we already know and can prove to be economically sound.

Highway Lighting To Reduce Accidents

DUSK is generally regarded as the most difficult time of day in which to drive. At dusk the driver does not see the details of objects, nor are there shadows and contrasts to assist in his quick perception of obstructions and holes in the road. Even though lights along the highway would not greatly increase the level of illumination, they would provide sufficient contrasts and shadows to assist the driver during this most difficult time of day.

At night, with the pupils of the driver's eyes open to their full extent and with the contrast of the black background, the light beam from modern head-lamps, even if properly adjusted, gives considerable glare; but not all head-lamps are modern and not all are correctly adjusted, and these lamps are a source of traffic accidents. With the highways properly lighted, cars could be driven virtually with daytime safety, with the beams dimmed or even using only the parking lights.

Where roadside pole lines exist, highway lighting units can be installed on these poles and provided with refractors that gather most of the light which otherwise would go upward and away from the road and direct it toward the road surface. With lamps of 600 cp., or 6000 lumens, in these fixtures, the maximum beam candlepower of each becomes approximately 10,800. A high level of illumination is not necessary to give quick perception on our highways; a motorist can see pedestrians and other objects with a comparatively low level of illumination by the silhouette and the glint effects.

A satisfactory highway-lighting installation could be made by spacing the lighting units 200 to 250 ft. apart at a mounting height of 25 to 30 ft. and using 6000-lumen lamps. The lamps would be arranged on a series circuit, and power would be collected from the power-distribution circuit at intervals along the line, possibly each four or five miles. The transformers could be controlled by a time switch or photoelectric cell, and the circuit then cascaded so that, when the first circuit was turned on, each circuit in succession would operate an oil switch in the primary of the next transformer and thus turn it on. The control would thus be cascaded progressively along the entire highway.

The total expense of such an installation probably would be $1,200 to $1,400 per year per mile, depending upon local conditions, including first cost, power and maintenance. The average economic loss of an automobile accident has been estimated at $1,000. If, with this highway lighting system, one accident per year per mile could be avoided, such lighting would cost the people of the State almost nothing. Fatal accidents, however, cannot be thought of as merely an economic loss; economic considerations are simply additional reasons for the installation.

Some day our principal highways will be lighted, and some agency will succeed in getting this done. Whatever agency does it will be giving the motorist as much for his money as the same amount expended in any other improvement. Organization and initiative will be required to accomplish this program.—From a Northwest Section paper by L. B. Robinson, of the General Electric Co., Seattle.

Late Developments in Airplane Stress-Analysis Methods and Their Effect on Airplane Structures

Aeronautic Meeting Paper

By Richard C. Gazley[1]

RECENT developments disclose the existence of a well-defined tendency toward greater accuracy and thoroughness in airplane stress-analysis methods, which serve only as a link between applied loads and allowable loads. This trend has just begun.

"Although we may justly look with pride on the aeronautical achievements thus far accomplished," the author says, "our knowledge and ability are far from being complete or entirely satisfactory." Hence, he analyzes several recurrent stress-analysis problems and indicates methods leading to their solution, because these seem to be outstanding in their ability to cause trouble for airplane designers.

Better understanding is needed of the peculiarities of aircraft structure; such as lack of rigidity, the nature of inertia loads, the effects of flutter and of engine vibration, and the dangers of stress concentration.

IT is probable that professional history will reveal no parallel to the rapid progress that has been demanded in the science of aeronautical engineering. This profession is young and its history short. The first flight of an airplane is within the short range of our own memories; and real progress in dealing with the complicated factors involved dates from a much later period.

Even now, although we may justly look with pride on the aeronautical achievements thus far accomplished, our knowledge and ability are far from being complete or entirely satisfactory. The exact nature and magnitude of the forces acting upon an airplane in flight, and certain of the materials used in aircraft construction, in the particular form and arrangement in which we must use them to accomplish the desired results, remain as partly unsolved problems. They will continue to engage our attention for some time to come.

Upon this academically tenuous foundation we are asked to erect a meticulous and exact science from which will result aircraft combining the somewhat paradoxical qualities of hitherto unachieved speed, the utmost in reliability and safety, and the most rigid economy in both operation costs and structural weight. And these demands are, of course, entirely just. There is no excuse for the continued existence of airplanes as commercial carriers unless they can be made to yield an adequate revenue with reliability, safety and comfort. It must also be remembered that their importance in the field of transportation is directly dependent upon their ability to travel at much greater speeds than those afforded by any other means of transportation.

In view of these facts it is indeed fortunate that our profession has attracted so many capable men and that the spirit of cooperation, of combining the efforts of a group toward the betterment of the entire industry, is so outstanding in the personnel of the aeronautical engineering profession. By means of this cooperation we are taking long strides in the desired direction. We are in the process of visualizing and reducing to formulas the exact loads which are imposed on an aircraft structure in service. We hope shortly to be able to predict with assurance the behavior of a specific airplane in all possible maneuvers. As new materials and methods of using them appear, we are gathering knowledge as to their behavior under various conditions so that we may know exactly the limits of their strength. A wealth of information on these subjects already is available and is steadily increasing in volume.

In the face of this increasing knowledge it becomes necessary for us to review our methods of structural design. Refinements of stress-analysis procedure were previously useless, and in many cases impossible, because of the approximate nature of the basic design conditions assumed and the lack of data upon which to base satisfactory conclusions. By refinements I am referring to more careful investigation of certain complex structural problems and more attention to important constructional details. Many such refinements are now not only possible but of paramount importance from the standpoint both of safety and the minimizing of useless dead weight. Many engineers have recognized this necessity and are working along the lines indicated. Others will follow, and it is expected that the subject eventually will assume the universal interest which it deserves.

It is believed, therefore, that the following discussion may prove of timely interest. Several recurrent stress-analysis problems are pointed out and methods leading to their solution are indicated. They have been chosen, not because they are the most important points in an airplane stress analysis, but because they seem to be outstanding in their ability to cause trouble for airplane designers.

Unsymmetrical Conditions

One of the most perplexing phases of stress-analysis procedure is introduced as a result of unsymmetrical loading conditions on the airplane. Use of the ailerons, flight through uneven gusts, landing on one wheel or

[1] M.S.A.E.—Chief, engineering section, aeronautics branch, Department of Commerce, City of Washington.

with some side drift, use of the rudder when the center of pressure of the vertical tail is above the fuselage—all of these conditions cause rolling or twisting stresses in the airplane structure, and it is important that their effects be predicted as accurately as possible. It is true that an arbitrary method of analysis has been established which seems to provide satisfactory structures in the case of conventional or ordinary airplanes. It is likewise true, however, that few of the aircraft being designed today are conventional or ordinary; the consequence is that special problems, outside the scope of previously established arbitrary rules, are beginning to arise with increasing frequency.

An example of common occurrence is the case of a multi-engine airplane being made to roll about its longitudinal axis by aerodynamic forces acting on the wings. Since air-transport operators are asking for every pound of payload that can safely be spared from the structure, it is necessary, economically, to take advantage of the fact that the engine nacelles play an important part in relieving the fuselage of some of the unsymmetrical stresses established. This can be accomplished in a fairly simple manner as follows:

The physical law for accelerated rotary motion is:

$$T = Iw$$

where

I = moment of inertia about the axis of rotation
T = unbalanced torque
w = angular acceleration

In engineering practice these quantities would be expressed most conveniently in foot-pound-radian-second units. The moment of inertia would, of course, be based on the mass instead of the weight of the airplane.

A general formula for the reaction of any given part of the airplane when subjected to angular acceleration can be developed about this fundamental formula, using the following symbols:

d_n = distance of the center of gravity of the part n from the axis of rotation, assumed as through the center of gravity of the airplane
F_n = reacting force due to component part n, in pounds
I_x = moment of inertia of the airplane about the X, or longitudinal axis
M = mass of the airplane, in slugs, $= (W/g)$
M_n = mass of the component part, in slugs, $= (W_n/g)$
T = unbalanced torque on the airplane, in foot-pounds
W = gross weight of the airplane, in pounds
W_n = weight of the component part n, in pounds
w = angular acceleration about the X-axis, in radians per second per second

The fundamental equation for rotary accelerated motion is, in this case

$$w = (T/I_x) \tag{1}$$

Since the angular acceleration w is the same for all parts of the airplane, the torque resisted by the part n is given by the equation

$$T_n = I_n w \tag{2}$$

The value of I_n is given very closely by the following equation:

$$I_n = M_n d_n^2 \tag{3}$$

Substituting Equation (3) in Equation (2), we have

$$T_n = M_n d_n^2 w$$

Replacing w by its value from Equation (1)

$$T_n = M_n d_n^2 (T/I_x) \tag{4}$$

The reacting force due to M_n is (T_n/d_n). Substituting for T_n from Equation (4) we obtain

$$F_n = M_n d_n (T/I_x) \tag{5}$$

If the moment of inertia, I_x, is known, Equation (5) gives the desired results.

To use Equation (5), I_x can be approximated by the method given in a report[2] entitled, Moments of Inertia of Several Airplanes, by Marvel P. Miller and Hartley A. Soulé. In this report, I_x is given by

$$I_x = C_a (W/g) (b^2 + h^2)$$

where

b = over-all span in feet
C_a = an empirical coefficient
h = over-all height in feet
W = gross weight in pounds

Substituting this value in Equation (5), we obtain

$$F_n = \frac{T}{C_a (W/g) (b^2 + h^2)} (M_n d_n) \tag{6}$$

If M_n is replaced by (W_n/g), Equation (6) becomes

$$F_n = \frac{T}{C_a W (b^2 + h^2)} (W_n d_n) \tag{7}$$

For a given design condition, the quantity $[T/(C_a W)(b^2+h^2)]$ is a constant and will be denoted by K. Then,

$$F_n = K W_n d_n \tag{8}$$

Values for C_a have been computed in N.A.C.A. Report No. 375 for several airplanes[2]. It would be desirable to use a value obtained from an airplane similar to the one being analyzed. In assuming a value for C_a, it should be noted that a high value will usually be conservative for flying conditions. The average value for the airplanes studied[2] is 0.0192.

From Equation (8) it is apparent that inertia forces due to unsymmetrical loading differ from those due to symmetrical loading only by the addition of the effect of the distance of the element of mass from the axis of rotation. It would, of course, be possible to perform an exact analysis for any unsymmetrical condition by computing the inertia load on each element of weight. This process would be quite tedious and the results usually would not warrant the extra labor. Consequently it has been customary to make conservative simplifying assumptions in analyzing the airplane for unsymmetrical loading. For example, the entire unbalanced wing load usually is assumed to be resisted only by the inertia of the fuselage and its contents. This makes it unnecessary to determine the effect of the wing mass and permits the direct application of the unbalanced reactions to the fuselage. Although this is conservative for the fuselage, it would be unwise to attempt to rationalize this procedure by accounting for the effect of the wing inertia until more data have been obtained on the probable unbalanced loads likely to be encountered in flight. The effect of engine nacelles is considerable, however, as has been indicated, and an investigation along these lines is well worth while in cases of that kind. Their reactions, as determined from Equation (8), can be applied at their centers of gravity as forces opposing the wing aerodynamic forces.

In dealing with the unsymmetrical flying conditions, it is convenient in most cases to separate the effects of direct loads from those due to torque loads. For example, if, in a monoplane, 70 per cent of the normal wing loading is assumed to be acting on one wing panel and 100 per cent on the other, the average load at the fuselage connections would be 85 per cent of the normal total load. Therefore the final unsymmetrical stresses can be obtained by adding the stresses due to 85 per cent of the normal loading to those obtained by considering 15 per cent of the total design load to be acting upward on one panel and 15 per cent to be acting downward on the other panel. By dealing separately with the torque

[2] See National Advisory Committee for Aeronautics Technical Note No. 375.

loads, it is possible to estimate the reactions produced by various items of weight, as already explained.

While on the subject of unsymmetrical wing loads, it might prove useful to discuss the case of a simple cantilever monoplane and the particular problem which it presents. In applying 70 per cent of the normal wing reactions on one side of the fuselage and 100 per cent on the other, it is necessary to remember that these reactions represent only the shear loads and that the unbalanced moment must be accounted for also. Where the spar is attached to the fuselage at only two points, there is only one possible way to represent this unbalanced moment; that is, by means of a vertical couple acting at the fuselage fittings. Where these fittings are close together, the forces due to this couple are likely to be much greater than the fitting loads obtained from any other design condition. Typical load, shear and moment curves for this condition are shown in Fig. 1. It is assumed that there is no beam load across the center section and that the wing fittings are comparatively close together. It should be noted that the couple required to balance 30 per cent of the total moment may cause one of the reactions to be upward, and that the other reaction is greatly in excess of the shear produced by the normal symmetrical design load.

Even though the correct forces on the fuselage due to the unsymmetrical conditions are known accurately, there remains the problem of distributing these forces properly through the fuselage structure. Since these conditions usually are critical only for a relatively small portion of the fuselage, it is customary to represent the remainder of the fuselage structure by applying arbitrarily assumed external loads to the portion under consideration. This practice should not be regarded as fundamentally sound unless the arbitrary external loads are based on a knowledge of the probable reactions which would be supplied by the missing part of the fuselage. On account of the difficulty of estimating the effect of certain portions of the fuselage, it is usually advisable to make what might be called "overlapping" assumptions. For instance, the assumption that part of the resistance to the rolling loads from the wing is supplied by the portion of the fuselage aft of the wing connections, and the remainder by the portion ahead of the wing connections, cannot possibly be conservative in both directions. It is therefore safer to design both the forward and rearward portions of the fuselage on the assumption that each resists more than one-half of the rolling loads, say 75 per cent.

It is most helpful, in making an analysis of this nature, to handle the rolling loads as was previously suggested; that is, to make the analysis for the forces due to 15 per cent of the design wing load acting upward on one side and downward on the other. The final stresses in the members can then be obtained by adding the stresses due to 85 per cent of the total load on the wings.

By dealing only with the rolling loads and drawing an isometric view of the structure, it is possible to determine the method of applying the resisting couples which will give the greatest loads in the bulkhead diagonals. A simple case is shown in Fig. 2. Here the wing reactions are represented in (*a*) by two couples *Fb* and *Rb*. It is assumed first that 75 per cent of the torque is resisted by the forward portion of the fuselage. This determines the couple *Ah*, which is applied so that bulkhead (1) must transform this couple into an equivalent couple acting on the side trusses. Fig. 2(*b*) shows the set-up for the design of bulkhead (1). Fig. 2(*d*) shows the loads finally applied to the side truss. For the opposite side, these loads would be reversed.

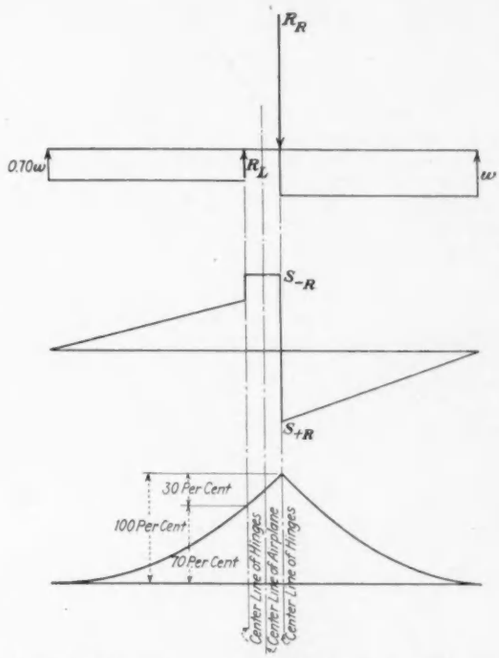


FIG. 1—UNSYMMETRICAL LOADS ON A CANTILEVER WING

The same process is carried out for the second assumption, in which 75 per cent of the total torque is carried rearward. The design loads for bulkhead (4) and the side truss are shown in Figs. 2(*c*) and 2(*e*). The side trusses shown in Figs. 2(*d*) and 2(*e*) are not individually in equilibrium. When the entire structure is considered, however, the unbalanced moment from one side truss is offset by an equal moment from the other, thus satisfying the conditions of equilibrium. The same general method can be applied to any type of airplane. Where it is possible to transmit torque through either the horizontal or the vertical trusses, it is generally conservative to apply all the load to vertical trusses, although the same principle of overlapping as-

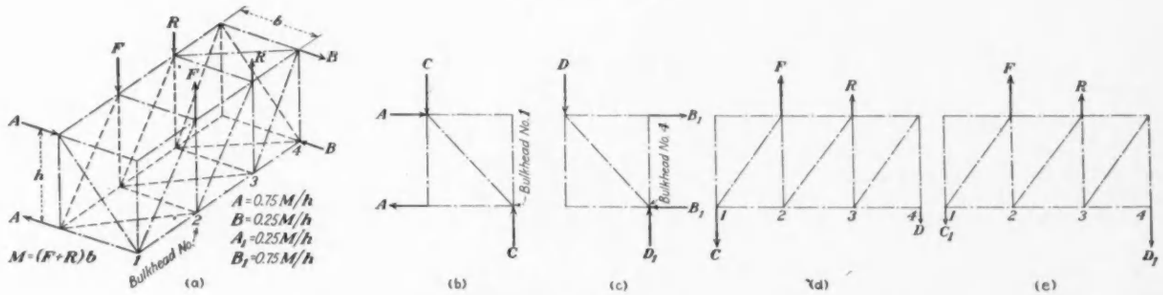


FIG. 2—UNSYMMETRICAL FORCES ON A FUSELAGE TRUSS

sumptions can be used to obtain loads in both sets of trusses.

The principles outlined for dealing with unsymmetrical conditions form the basis for the semi-empirical method now recommended by the Department of Commerce for analyzing the fuselage for fin and rudder loads. In this method the assumption that one-half the torque is resisted by the top and bottom trusses is believed to be somewhat conservative, due to the greater strength of the side trusses on almost all airplanes. The results are, however, much more favorable to the airplane than the formerly usual method of completely neglecting the effect of the side trusses. On large airplanes having high twisting moments produced by the vertical-surface forces, the reduction in design stresses for the top and bottom trusses and the consequent reduction of structural weight may be of considerable importance.

The determination of the bulkhead loads is based on the fact that certain bulkheads, on account of their location in the fuselage structure, must transmit the total couple existing in one set of trusses into the other set, while the remaining bulkheads must transmit only relatively small portions of the couples in one pair of trusses into the other. For instance, if the diagonal bracing in one bay of the top truss is omitted, the top truss at this point becomes incapable of transmitting a shear load between adjacent stations. Consequently the shear load must be transferred to the bottom truss by means of the bulkheads aft of the open bay. If the nearest bulkhead is capable of transmitting all the shear load, the ones farther aft are not necessary and therefore do not require to be highly stressed. More exact methods of analysis have shown that the actual loads in the intermediate bulkhead bracing are usually quite small, and that the stiffening effect of those bulkheads is much less than might be supposed.

By treating the question of twisting moments on the fuselage in a more rational manner, several interesting facts, which might not otherwise have been readily perceived, have become apparent. Two of these have already been discussed; that is, the relative unimportance of intermediate bulkheads and the effect of an unbraced bay on the rest of the structure. Another is the possibility of saving weight by omitting entirely the diagonal bracing in the bottom or top truss, in which case the side trusses would be made to carry the total twisting moment. A fourth and rather unusual proposal is the feasibility of leaving all diagonal bracing out of a side-truss bay. There is no reason why each side truss must be considered separately, and it would be entirely possible to arrange a system of bulkheads so that the shear load from one side truss could be transmitted across the fuselage into the other and back again at a convenient point. In this way the top and bottom truss bracing could be utilized to resist the couples applied by the bulkheads, and the usual awkward arrangement of heavy brace-tubes in one corner of the side bay could be dispensed with. To the best of my knowledge this type of construction has not been used deliberately on any airplane, although probably many cases exist in which the structure actually has benefited by the operation of this principle.

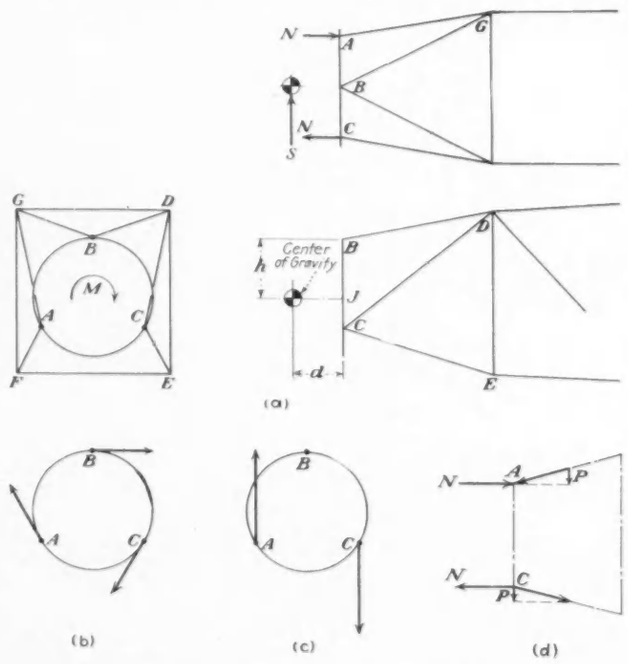


FIG. 3—ENGINE MOUNT WITH THREE SUPPORTS

One more practical example might be cited to demonstrate a typical three-dimensional problem in unsymmetrical loading. Fig. 3 shows an engine mount having three points of support for the mounting ring. These three points form the apexes of three triangles, two of which lie in vertical planes while the third is a part of the top truss. By simply following the usual method of distributing the torque load, we would obtain equal tangential force vectors at points A, B and C, as shown in Fig. 3(b). But, in the type of construction shown, there is no bracing to prevent either point A or point C from moving sidewise. Therefore they are unable to resist forces having side components. The result is that point B is not effective in resisting the torque load, since there can be no unbalanced side components. The torque load must therefore be applied as a couple at points A and C, as shown in Fig. 3(c).

The same principles can be applied to the analysis of this structure for side load. Here an unusual condition exists on account of the side truss-members not being parallel in the top view. If we start by moving force S rearward to point J, the corresponding moment Sd must be applied. This must be resisted entirely by fore-and-aft loads in the side trusses. If Sd is represented by a couple consisting of forces N applied in the top view, it can be seen from Fig. 3(d) that the structure can resist side loads of magnitude P at points A and C. Therefore point B will have to resist only a side load equal to $(S - 2P)$. A vertical couple at points A and C will be required to represent the difference of the moments produced by moving force $2P$ from point J down to point C and force $(S - 2P)$ from point J up to joint B.

A simple method for determining the amount of side load which will be taken at points A, B and C is shown in Fig. 4. In the top view the lines representing the planes of the side trusses are extended to their point of intersection, Q. The side load resisted by point B will be equal to (b/c) S. The total side load resisted by points A and C will equal (a/c) S. The vertical couple to be resisted at points A and C will be equivalent to the moment $M = S\,[(b/c)m - (a/c)n]$. The horizontal couple at points A and C will be equivalent to Sa.

Wing-Spar Analysis

In the case of externally braced wings, some uncertainty often exists as to the actual conditions in the region adjoining the strut point. A simple case is illustrated in Fig. 5. In analyzing the spar, the first step would be to compute the moment curve, considering point A as the strut point. This would determine the correct curve for the portion of the spar inboard of the actual attachment point. The overhang-moment curve would then be extended to point B. This portion of the curve determines the moments outboard of the ac-

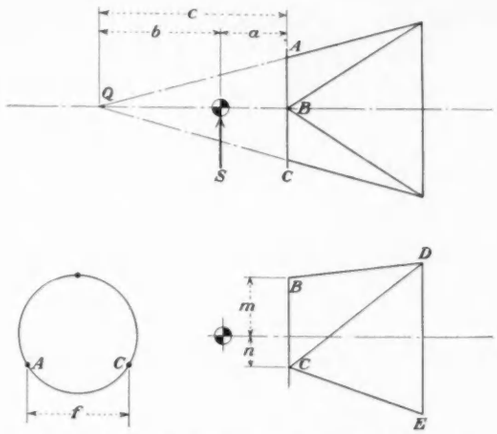


FIG. 4—SIDE-LOAD ANALYSIS FOR AN ENGINE MOUNT WITH THREE POINTS OF SUPPORT

tual attachment point. The moment represented by the distance *B-C* on the moment curve should equal the moment produced by the component of the strut load parallel to the spar axis acting about the neutral axis of the spar.

For truss-type spars, it often is advisable to isolate a portion of the spar in the region of the strut point and apply the various loads produced by the brace members, moments, shears and local air loads. The structure thus isolated usually can be analyzed graphically after making a few simplifying assumptions. Fig. 6 illustrates such a condition, in which the forces are represented as follows:

A = air load from rib
F_i = chord couple due to inboard moment
F_o = chord couple due to overhang moment
R = resultant brace-member load
S_i = inboard shear
S_o = outboard shear

The most difficult part of a problem of this nature is the proper application of the force R. In this example the force might be divided between stations (1) and (2) by the usual method of analyzing fittings subjected to moments, which is more fully discussed elsewhere in this paper. The final loads at stations (1) and (2) would therefore be found by dividing the load R equally between the stations and applying a couple equivalent to the moment of R about the mid-point between stations (1) and (2). The resultant loads are shown in Fig. 7. Having determined all the external loads acting on the structure, the loads in the chord and web members can be found by the usual graphical method.

In connection with wing-spar analysis, it should be pointed out that certain designs will admit of considerable deflection at the point of external support. This is particularly true of wire-braced monoplanes having relatively small angles between the wires and spars. To show what can be done along these lines, a simple case such as that shown in Fig. 8 will be treated in a general manner. Here a uniformly distributed load is assumed in each bay, dihedral is neglected, and a symmetrical loading is used. The structure is indeterminate because of the continuity of the spar at points (1) and (2).

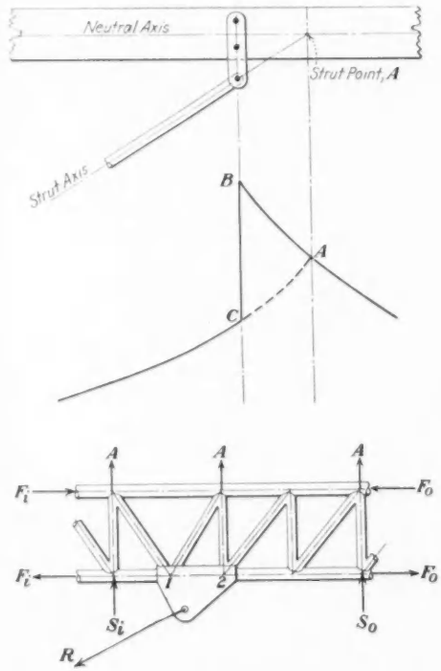


FIG. 5—STRUT POINT ON A WING SPAR

FIG. 6—WING-SPAR ANALYSIS

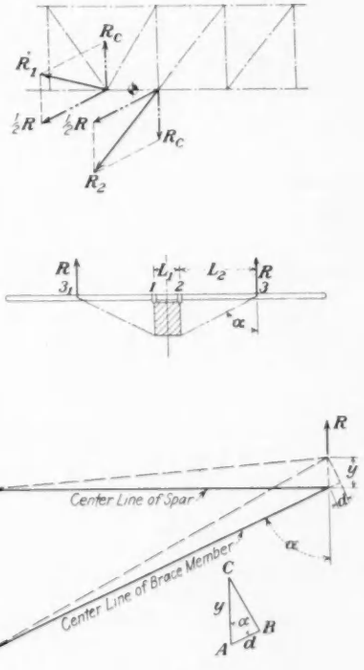


FIG. 7—DIVISION OF STRUT LOAD ON FITTING

FIG. 8—SYMMETRICALLY LOADED INDETERMINATE SPAR

FIG. 9—DEFLECTION AT BRACE POINT ON A WING SPAR

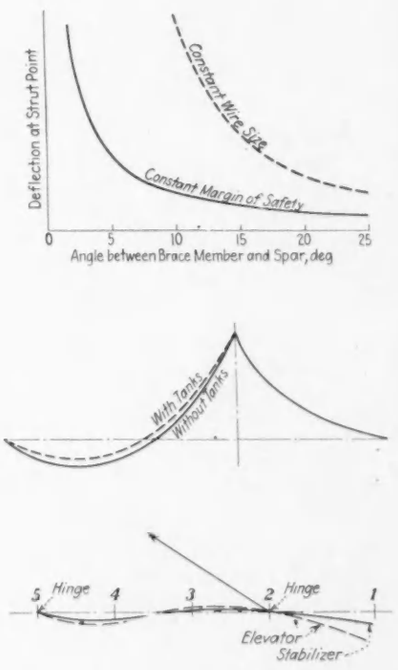


FIG. 10—DEFLECTION OF A SPAR AT A STRUT POINT

FIG. 11—BENDING-MOMENT CURVE FOR A SPAR WITH FUEL TANKS LOCATED IN WING

FIG. 12—DEFLECTION OF TAIL-SURFACE SPARS

The general method of handling this problem would be to consider the points of support to be in line and to write two three-moment equations to determine the moment at (2). By assuming an unknown deflection y to exist at (3) and (3_1) and writing the three-moment equation for one support out of line, the following simple formula is obtained by making use of the fact that the moments at points (1) and (2) are equal:

$$M_3L_2 + M_2(3L_1 + 2L_2) = \frac{w_1L_1^3}{4} + \frac{w_2L_2^3}{4} + 6\left(\frac{EI_2y}{L_2}\right) \quad (9)$$

where

E = modulus of elasticity of spar
I = moment of inertia of spar
L = length of bay
M = bending moment
w = uniform loading per unit of length
y = deflection at point (3).

It is apparent that this formula enables M_2 to be determined immediately if the supports are assumed to remain in line; that is, if y is assumed equal to zero. When the brace member is a wire and where the angle α is large, it is well, however, to investigate the effects of deflection at the external support.

Fig. 9 shows an exaggerated picture of the deflections at the external brace point. If the actual deflections are small compared with the lengths of the brace member and spar, the figure $A\ B\ C$ can be considered to be a right triangle having one angle equal to α. From this triangle

$$y = (d/\cos\alpha) \quad (10)$$

Since the spar and the brace member are in the same vertical plane, the load in the brace member is easily determined, since

$$P = (R/\cos\alpha) \quad (11)$$

But

$$d = \frac{PL_b}{A_bE_b} = \frac{RL_b}{A_bE_b\cos\alpha} \quad (12)$$

where

A_b = area of the brace member
E_b = modulus of elasticity of the brace member
L_b = length of the brace member

Substituting Equation (12) in Equation (10), we have

$$y = \left(\frac{RL_b}{A_bE_b}\right)\left(\frac{1}{\cos^2\alpha}\right) \quad (13)$$

The importance of the angle α should be noted. As α approaches 90 deg., that is, as the angle between the brace member and the spar is decreased, cosine α approaches zero and the deflection y approaches infinity. The fact that cosine α is squared in the expression for y greatly magnifies the effect of small angles between the spar and brace member.

Before returning to the problem of the wing spar, it is of some interest to note that, if the unit stress in the brace member is to be held constant, Equation (12) becomes $d = fL_b/E_b$ and Equation (13) may be written

$$y = fL_b/E_b\cos\alpha \quad (14)$$

where f = unit stress in the brace member.

Equation (14) indicates that, even though the brace-member size is varied so as to maintain a constant margin of safety, the deflection will be inversely proportional to the cosine of the angle α. Curves illustrating the dependence of the deflection on the angle between the brace member and spar are shown in Fig. 10. It should be noted that a constant margin of safety is assumed for one of the curves.

Returning now to our original consideration of the structure shown in Fig. 8, it appears that the term y, representing deflection at the strut point, might be replaced in Equation (9) by some function of the bending moments and spar properties, thus enabling a solution to be made for M_2.

The value of R can be found from the equation

$$R = S_{+3} - \left[\frac{-M_2 + M_3 + \left(\frac{wL_2^2}{2}\right)}{L_2}\right] \quad (15)$$

where S_{+3} is the shear just outboard of station (3). By substituting the foregoing value of R in Equation (13) and the resulting value of y in Equation (9) and simplifying, the expression obtained for M_2 is

$$M_2 = \frac{\left(\frac{w_1L_1^3}{4}\right) + \left(\frac{w_2L_2^3}{4}\right) - M_3L_2 - K\left[L_2S_{+3} - M_3 - \left(\frac{wL_2^2}{2}\right)\right]}{3L_1 + 2L_2 + K} \quad (16)$$

where

A_b = area of brace member
E_b = modulus of elasticity of brace member
E_s = modulus of elasticity of spar
$K = 6\left[(L_bE_sI_2)/(L_2^2E_bA_b\cos^2\alpha)\right]$
L_b = length of brace member
α = angle between brace member and vertical, and the remaining terms have their usual significance

In Equation (16) all the terms involving the physical properties of the structure are combined into a single constant, leaving only the usual known quantities in the expression for M_2. Similar equations could be derived for various other conditions, such as the case where one moment is zero; structures not lying in a vertical plane; spars having non-uniform or concentrated loads; and the like. Since the secondary bending due to axial loads is not likely to cause much change in the values of moment at the points of support, it is probable that a close approximation would be obtained by Equation (16), which could be rechecked for the effects of secondary bending after the brace-member load had been determined. Equation (16) might also be applied to certain types of tail-surface structure.

In considering the effect of placing the fuel tank in the inner bay of an externally braced wing, one is tempted to assume that the bending moment is decreased and that the effect is to reduce the spar stresses. Referring to Fig. 11, the full line represents the original moment curve, and the dotted line the curve after the effect of the dead weight is considered. It is obvious that the effect is to increase the bending moments between the strut point and the point of inflection. This section often contains a critical point where the strut-point reinforcement is discontinued, and it is therefore necessary to investigate the effect of weights of this kind.

Tail-Surface Analysis Based on Deflections

Some of the methods of handling indeterminate structures already have been demonstrated for particular cases. One of these methods, because of its usefulness, deserves separate attention; that is, the method which might be termed one of "consistent deflections." It is shown elsewhere in this paper that many statically indeterminate wing spars, built-up fittings and landing-gear members can be analyzed readily by making use of the fact that the deflections of two or more parts must be equal. This principle can be summarized as follows: The division of load between two or more statically indeterminate members will be in proportion to the loads required to produce unit deflections in each of the members. This method is a particular simplified case of the broader method of least work and is much easier to apply when conditions permit its use.

The limitations of the three-moment equation already

have been mentioned; but, to emphasize this point, it may be well to discuss a typical case with reference to tail surfaces. An analysis of the elevator and rear-stabilizer spars is often based on the assumption that the elevator hinges remain in a straight line. The effect of this untenable assumption is to relieve the bending stress in the outer portion of the elevator spar, increase it in the corresponding portion of the stabilizer spar, and give fictitious loads at the hinges. Anyone who has seen a static test of a tail-surface structure has noticed that the outer end of the stabilizer spar deflects downward considerably, and that the elevator and stabilizer spars deflect more or less together, depending on the number of hinges. Consequently, if assumptions are to be made, it would be more logical to assume that both spars deflect together under the combined loading for the two, and that the moment is divided between the two spars in proportion to their moments of inertia and moduli of elasticity.

If we consider a case such as that shown in Fig. 12, where there are hinges only at points (2) and (5), it is obvious that each spar would deflect independently and that the hinge at the strut point would be subjected to the reaction from the elevator spar at that point. The neutral axes of the spars might have a relative displacement such as that shown exaggerated in Fig. 12. The addition of hinges at points (1), (3) and (4) would bring the neutral axes into approximate agreement, the amount of load taken by each hinge depending to a large extent on the original displacement between the neutral axes at the point in question. If the original deflected positions of the spars happened to be very nearly the same, it is obvious that the hinge at (2) would take most of the load, regardless of the additional number of hinges used. It is conceivable that some form of loading might exist, not accounted for in the design loads, such that the spars would deflect nearly together and cause the hinges nearest the points of support to take most of the load. Therefore it would be well to assume that the hinges nearest the main points of support must be capable of transmitting the elevator-spar reactions computed for only two points of support. The load on any other hinge might be conservatively estimated to be equal to the load acting on the portion of the elevator area in which the hinge lies.

Where the spars are continuous across the fuselage, the same principles can be applied, assuming that the spars act as a single spar if there are numerous hinges and that the hinges near the points of support must transmit most of the reactions. Where the outer point of support is subject to considerable deflection, the effect of this deflection can be accounted for by the method proposed in this paper under the subject of Wing-Spar Analysis.

For internally braced structures, the same line of reasoning will show that, unless the deflection of the two spars happens to be very nearly the same, the outer hinge will take most of the load. For instance, the weaker the elevator spar is in proportion to the stabilizer spar, the more load will be taken by the outer hinge. If the elevator spar were supported only at the inner and outer ends of the stabilizer spar, the outer-hinge reaction would be the maximum obtainable. This value could be used as a conservative estimate of the outer-hinge reaction for any case in which the elevator spar is relatively weak or discontinuous across the fuselage.

In any of the foregoing cases it would, of course, be possible to make a precise analysis based on the fact that the deflection at the hinges must be equal. However, the use of conservative assumptions based on a clear conception of the conditions will often eliminate a great deal of long and meticulous computation. In any event, it is better to make an intelligent approximate analysis than to obtain misleading results by the use of a precise method based entirely on conditions which are not fulfilled by the structure.

Landing-Gear Analysis

The next subject with which we have to deal presents a series of complex structural-analysis problems which is perhaps unequaled in any other part of an airplane. I refer to the landing gear; and this series of complexities is rather surprising at first glance because of the fact that many landing-gear structures consist of only three members on each side, in addition, of course, to the wheels and tires, which are handled separately as distinct units.

The necessity for concentrating attention on this unit is illustrated by our commercial-airplane accident records, which show that, although all other types of structural failure have been reduced to a very small figure, landing-gear failures continue to be almost common occurrences. There are, of course, many contributory reasons for this which are outside the control of the designer; but, without doubt, structural design plays an important part in this record. That this condition has been realized is indicated by the many excellently comprehensive chassis analyses which have recently been produced, but among these there is very little agreement as to the general method followed, particularly with respect to conventions and the manner of distributing the effects of moments. It is believed, therefore, to be worth while to point out what seem to be the most satisfactory conventions, assumptions and methods of attack leading to the uniformity and clarity which we all desire.

As a beginning, it seems desirable to adopt a uniform system of reference axes, such as those standardized by the National Advisory Committee for Aeronautics. In this system, positive directions and forces would be measured rearward, outward and upward along the X, Y and Z-axes respectively. These same axes can be used in dealing with moments, by referring any moment to the axis about which it acts. For instance, following the N.A.C.A. convention for positive moments, it would be necessary to use only the "left-hand rule," in which the direction of the moment is indicated by the fingers of the left hand when the thumb points in the direction of the vector. The axis would, of course, be determined by the vector itself. Various forms of moment vectors have been devised, one of the most common being the use of a double arrow to indicate a moment, the single arrow being used to indicate direct forces.

To illustrate more clearly the use of moment vectors, if the moment due to an upward Z-force acting on the wheel is considered, its axis, magnitude and direction could be represented by a double-headed arrow pointing rearward, parallel to the X-axis. Similarly, any combination of moments acting about a point can be resolved into a single resultant moment by any method which can be used for direct forces.

Another well-known convention which eliminates a great deal of confusion is the initial assumption that all unknown loads are tension and that a negative load indicates compression. Any attempt to predict the nature of the internal loads before making the analysis is entirely unnecessary and quite confusing.

By far the most troublesome part of a chassis analysis is the determination of the stresses due to overhang and brake moments. These moments have two major effects; that is, they affect the axial loads in the members and add torsion and bending stresses. The effect on the axial loads can properly be accounted for only after something is known about the manner in which the moments are resisted. Due to the wide variety of chassis designs, it is impossible to devise a general method for handling moments which would be sufficiently simple for practical use. Several suggestions might be made, how-

ever, to show the possibilities of reasonably accurate treatment.

By dealing with moment vectors it is possible to resolve the external moments into other convenient systems of independent axes. For example, if only one member is capable of resisting torsion and bending, the problem is easily solved by transforming the external moments into three equivalent moments acting about axes respectively coincident with and perpendicular to the member in question. Usually, however, two or more members are capable of taking bending and torsion. The structure is then indeterminate with respect to moments and some method based on deflections, work energy or intelligent assumptions must be employed. In cases where a V-structure must resist all the moments, it is convenient to resolve the external moments into equivalent moments acting about axes lying perpendicular to and in the plane of the V-structure. The moment about an axis perpendicular to the V-plane usually is divided between the tubes in bending in proportion to their moments of inertia and in inverse proportion to their lengths. This would be correct for two tubes welded together at one end and incapable of taking bending at the other ends, a condition usually approximated in practice.

The distribution of moments whose axes lie in the plane of the V-struts requires the application of some method involving "least-work" or strain energy. In many cases, however, an inspection of the conditions will indicate the method by which most of the moment is likely to be resisted, and it is therefore possible to make reasonable simplifying assumptions. A great deal of information can be obtained by drawing the V-structure in true view, applying the moment vectors and determining the resultant moment.

To illustrate the above principle graphically, external force vectors and moment vectors have been applied to a conventional structure in Fig. 13. In this case a universal-joint is assumed at each end of the shock strut

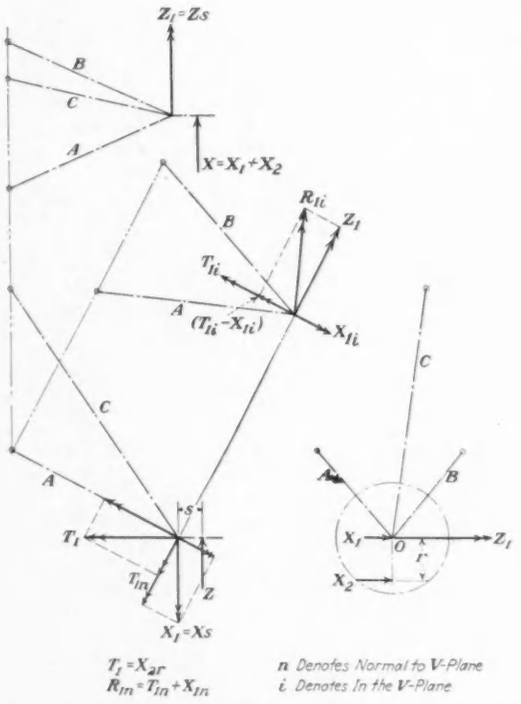


FIG. 13—FORCE VECTORS AND MOMENT VECTORS FOR LANDING-GEAR ANALYSIS

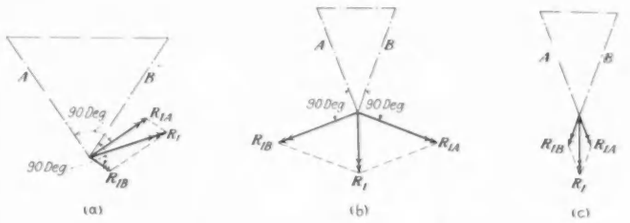


FIG. 14—RESOLUTION OF MOMENT VECTORS

and at the inner ends of the V-members. The applied loads—which have been resolved into components along the reference axes—are shown by single-arrowed vectors and designated by letters referring to the components shown. The same letters, with subscripts, are used to denote the moments, which are shown by double-arrowed vectors indicating the axes, direction and magnitude of the moments. Each direct force is moved to the origin, *O*, and the corresponding moment vector is applied at that point. As the shock strut is unable to resist either bending or torsion, all the moment vectors must be applied to the V-structure. The resolution of these vectors into resultant-moment vectors parallel and perpendicular to the V-plane is shown graphically in Fig. 13.

By examining the projected view of the V-structure, it is possible to estimate the effect of the resultant moment on the two members. If they were attached at the fuselage by ball-and-socket joints incapable of resisting torsion, the resultant R_1 lying in the plane would have to be resisted entirely by bending. The bending moments would then be determinate and are shown graphically in Fig. 14(*a*). In some cases it would be logical to assume such a condition. On the other hand, if the angle between V-members happened to be small and if R_1 should lie between the members or their projections, the bending stresses required to resist R_1 would be excessively high, as shown in Fig. 14(*b*). This shows that, if the members *A* and *B* were capable of resisting torsion, they would be more effective in torsion than in bending. It might therefore be logical to assume all the moment R_1 to be resisted in torsion, as shown in Fig. 14(*c*).

The foregoing discussion indicates the possibility of making reasonable arbitrary distributions of moment without going into involved mathematical computations. In every case the relative strengths of the members, as well as their position and manner of attachment, need to be considered. In general, the use of intelligent assumptions should introduce errors no greater than those to be expected in any indeterminate analysis in which the physical properties of the members play an important part.

The effect of moments on the axial loads can be determined by finding the forces required at the origin to balance the net fuselage reactions due to the moments. These forces can be applied as external loads and the resulting internal axial loads can then be found by the usual method. Or, alternatively, the final axial loads can be determined in one operation by applying the design loads and the moment reactions simultaneously.

Fitting Analysis

The orphans of an airplane structure, insofar as stress-analysis attention is concerned, have until recently been the fittings which connect the important primary members to one another. This is sometimes difficult to understand, because good fittings usually are not easy to design and even the most innocent-appearing ones develop unexpected weaknesses. The science of fitting design is, in reality, quite complex and is, in its entirety, too comprehensive for the scope of this paper. Certain

of the outstanding phases, however, are worthy of discussion.

Several causes of fitting failure should be recognized so that they can be avoided so far as possible. Among these is the tendency to regard a fitting as a minor part of the structure and to be content with superficial methods of stress analysis. Other causes are the effects of fabrication methods, eccentricities, stress concentration, vibration and fatigue. A common source of trouble in the analysis of a bolted or riveted fitting is the effect of a moment applied to the fitting by eccentric loading or some other means. Although the correct procedure in this case is fairly well known, it will be reviewed briefly on account of its importance.

Any plane system of loads on a fitting can be resolved into a resultant load acting at the centroid of the fitting bolts and a resultant moment acting about the centroid of the bolt areas. In Fig. 15(*a*) the conditions produced by the forces H and V can be represented by the resultant R acting at the centroid of the bolts and the moment Rd tending to twist the fitting around the centroid. It is first assumed that R is divided between the bolts in proportion to their cross-sectional areas. The moment Rd is then divided between bolts so that the moment forces on the bolts are in proportion to their areas and distances from the centroid. The direct forces due to R will, of course, act parallel to R, while each force due to the moment will make an angle of 90 deg. with the radius line from the centroid to the bolt in question. The final load on each bolt will be the vector sum of the two loads acting on the bolt. The load vectors which would be obtained for the arrangement under consideration are illustrated in Fig. 15(*b*).

When a graphical solution is performed, it is possible to determine the centroid of the bolt areas very quickly by the following method. Assuming bolts of equal size in Fig. 15(*c*), a line is first drawn between any two bolts and bisected. The point so determined is joined to another bolt center and the resulting line is divided one-third the distance from the end first determined. The process is repeated until all the bolts have been included, the divisor for the line being in each case the number of bolts which have been connected. If the bolts are of different size, the division should be in the ratio of the bolt area last considered to the total bolt area connected by the lines.

Recent practice in constructing built-up fittings by the welding together of component parts has led to designs in which the welding area is reduced to the minimum. It is therefore necessary to investigate the actual strength of the welds to determine the total fitting strength. In many cases failure will occur by a combination of different types of failure, and this condition must be foreseen by the designer. Without going into the allowable unit stresses which can be used safely for welded joints, I shall attempt to bring out a few of the

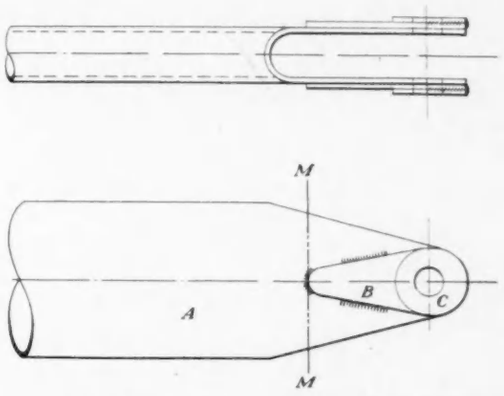


FIG. 16—TYPICAL WELDED STRUT-END FITTING

points which are involved in determining the applied stresses on various component parts of a welded fitting.

The most reasonable assumption which can be used as a basis for fitting analysis is that there is no bending in the connecting bolt or pin and that, consequently, the deformation of the fitting is uniform over the cross-section in contact with the bolt. The amount of load taken by any component part of the fitting will therefore be proportional to its area of contact and the load required to produce a unit deformation in the direction of the applied load. This principle can be applied to many types of fittings, including those made up of widely different kinds of material.

When the component parts of a fitting are of the same material and a standard bolt or pin is used, the load taken by each part usually will be proportional to the thickness of the part. This assumption can be made for the fitting shown in Fig. 16, which will be used as an example. In this fitting, parts C and B are welded together only around the end, while part B is welded to A at several points behind the bolt hole. As the applied load in this case is symmetrically located, we may consider only one side of the fitting, composed of parts A, B and C. The allowable bearing load would be computed as usual, considering the whole bearing area to be effective. The loads taken by parts A, B and C can be found by considering the relative thickness of the parts. The load taken by washer C must obviously be transmitted to part B through the edge welding. There is no possibility of part C failing in tension or shear, so the welding strength is critical.

Part B must transmit the combined loads taken by B and C into A through the welding behind the bolt hole. This probably will be the critical condition; but an investigation of part B for tension across the bolt hole should be made, using the combined loads from B

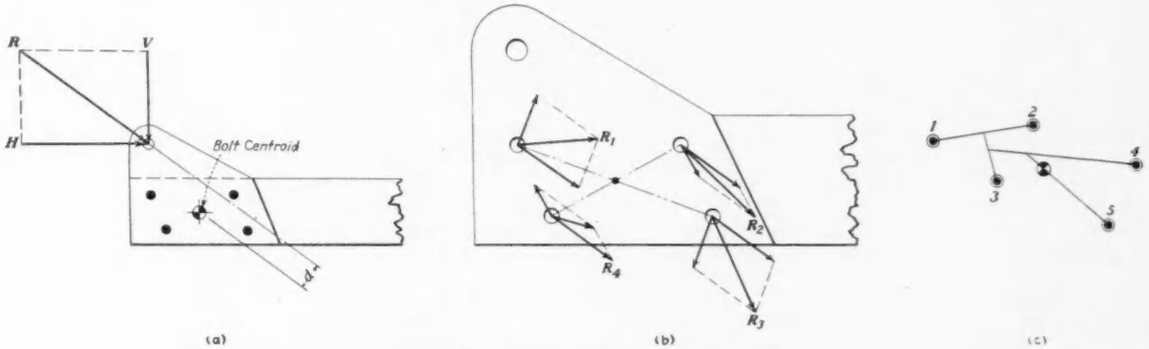


FIG. 15—ECCENTRIC FORCE ACTING ON A FITTING

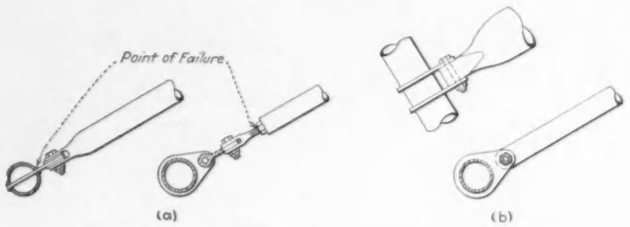


FIG. 17—TYPES OF STRUT-END FITTINGS

and *C*. The possibility of shear failure by "tearing-out" should be investigated also, although this form of failure is difficult to analyze properly. In this case it would be best to investigate the ratio between the combined strength of the weld plus the allowable shear through part *B*, and the total load carried by parts *B* and *C*. Part *A* is subject to failure by tension or shear under the load which it receives from the bolt. Section *M-M* will, of course, have to carry the entire load in tension.

The investigation of this fitting for compression loads would be made similarly, except that part *C* would then be subject to failure in tension and shear. It might be well to note that the customary method of investigating shear or tear-out conditions is to use twice the area cut by a section through the line of pull, rather than the area cut by parallel sections tangent to the bolt hole.

From the example just considered, it is obvious that there is a possibility of progressive failure of fittings made up of component parts. That is, a condition may exist in which the sum of the strengths of the various parts is greater than the design load but the design is such that the total allowable loads for all parts cannot be realized simultaneously.

The effects of fabrication and assembly methods are beyond the scope of this paper. A word may be said, however, about the effects of vibration. It has been noticed that the fittings used to attach exposed wires or small streamlined tubes often break off at a point having a very high apparent margin of safety. In nearly every case of this nature it has been found that the pin or bolt was perpendicular to the major axis of the strut or wire, as shown in Fig. 17(*a*). Even a small amount of vibration in the strut would cause bending at the root of the fitting plate. The fiber stresses due to bending would add to the direct stress and would also vary rapidly during the vibration of the strut, eventually causing failure at a direct load considerably less than the design load. The remedy which suggests itself for this situation is shown in Fig. 17(*b*), where the pin is placed parallel to the major axis of the strut. This type of design also eliminates an objectionable abrupt diminution in cross-section between the lug and the supporting structure.

Jury Struts

Among designers of strut-braced monoplanes there is an increasing tendency to reduce the weight of the external wing-bracing by providing the main struts with lateral support at about one-third the distance in from the outer ends. This support is furnished by a small auxiliary strut, commonly called a jury strut, which is attached to the wing spar at its upper end. This type of design is quite effective for the purpose intended, but it has introduced some difficult analysis problems.

The case in which the auxiliary strut and upper end of the main strut are both pinned at their intersection is fairly simple and has been successfully analyzed by a number of designers. The more common case, however, where the main lift-strut is continuous, is greatly complicated by the number of factors affecting the force distribution. A precise solution of this problem probably would result in unwieldy formulas but would enable the importance of the various factors to be determined. As there exists, to my knowledge, no complete analysis of the effects of an auxiliary strut on the spar and main strut, it might be useful to consider a few of the more important principles.

It is sometimes stated that the effect of the auxiliary strut depends largely on its location with respect to the point of inflexion of the spar; that is, where the bending moment is zero. It would be more nearly true to say that the critical point in this respect is the point where the spar deflection is zero with respect to a line joining the two points of support. This point can be determined by writing the general expression for the deflection at any point and setting it equal to zero. It can be seen readily that the point of inflexion and the point of zero deflection may be quite different. The significance of the latter point is shown in Fig. 18, where the broken line represents the original position of the neutral axis. A downward load is assumed, as we are concerned chiefly with the column loading of the main brace-strut.

In the case illustrated, the large overhang causes the point of zero deflection to be well in toward the root of the spar. The point of inflexion is outboard of the point of zero deflection. Center lines for an external brace-strut and auxiliary strut have been included. Although the diagram is exaggerated, it is easy to see that, instead of acting as a stable point of support, the auxiliary strut will produce bending in the main strut, due to the upward deflection of the spar. The axial compressive load in the main strut will produce additional secondary bending due to the deflection caused by the auxiliary strut. The action will multiply itself until a state of equilibrium is reached, possibly after the buckling action of the lift strut has caused the auxiliary strut actually to push up on the wing spar, thus increasing the spar bending-moment at that point. As in the case of a simple column, there will be a critical axial load beyond which excessive lateral deflection of the strut will take place.

From these considerations we can at least infer that auxiliary struts may, in some conditions, have unforeseen harmful effects on the brace strut, the wing spar, or both. The usual practice of assuming a restraint coefficient greater than unity at the point of attachment of the auxiliary strut is highly questionable. In some cases it is even quite probable that analyzing the strut as a pin-ended column between points of support is not severe enough.

As I have already stated, the precise solution of this problem is at present beyond the range of practical application and offers an interesting challenge to those who may be looking for more advanced forms of investigation. To avoid a dangerous situation, however, it is necessary to give careful consideration to the peculiarities of each case of this nature. For example, the point of zero deflection can be determined approximately and used for the location of the auxiliary strut, if this is

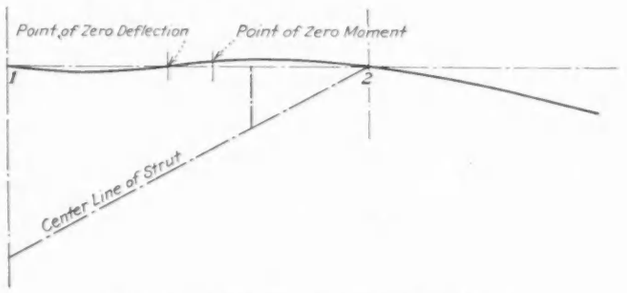


FIG. 18—WING-SPAR DEFLECTION CURVE

practicable. If not, a rough idea of the condition can be obtained by finding the unrestrained deflection of the spar at the auxiliary-strut attachment point.

The design of the main lift-strut then resolves itself into the problem of finding the critical load for a pin-ended long column initially straight but deflected laterally a constant amount at some point along its length. To be more accurate, the deflection could be taken as a linear function of the axial load in the column. The solution of this problem would be a valuable addition to our knowledge of strength of materials. Pending such a solution, we must rely on empirical formulas and meager test data for our allowable loads and therefore need to provide ample margins of safety.

It is, of course, possible to determine the maximum unit stress in the brace strut by means of the precise equations for an auxiliary loaded beam with supports out of line. This would permit the calculation of a margin of safety based on the yield point of the material, but would not indicate how close to the critical axial load the design axial load approached. In other words, when dealing with long columns, we are interested mainly in the magnitude of the axial load which will produce elastic instability, as that load is usually the maximum allowable load.

General Considerations

In conclusion I should like to point out that the most outstanding fact disclosed by a review of recent developments is the existence of a well-defined tendency toward greater accuracy and thoroughness in stress-analysis investigations. This trend, however, is only at its beginning and there still remains much ground to be covered, as has been indicated by certain of the cases cited in this paper. We need a fuller understanding of some of the peculiarities of aircraft structure; as, for example, their lack of rigidity, the nature of inertia loads, the effects of flutter and engine vibration, the dangers of stress concentration, and the like. Without such an understanding it is difficult to obtain, by the conventional arbitrary methods of stress analysis, a truly efficient structure; that is, a structure of uniform strength and minimum weight.

Most of the foregoing principles have been demonstrated in this paper by practical examples. The subjects of stress concentration and vibration effects are not properly included under the general heading of stress-analysis methods, but they are of such importance that every designer needs to become acquainted with the principles involved. Both of these subjects have been covered by various authors in the past, and an excellent treatment has recently been developed by S. Timoshenko, in which quantitative data are presented.

In considering stress-analysis methods, it always should be borne in mind that they serve only as a link between applied loads and allowable loads. The question of what the applied loads should be is by no means solved, and in many cases of combined stresses there still exists some measure of uncertainty as to what the allowable loads should be. To be inaccurate in our methods of stress analysis would therefore amount to broadening the range of the probable errors involved in the complete process of determining the airworthiness of an airplane. From this viewpoint, the importance of proper and careful stress analysis becomes evident.

Importance of Aviation in Canada

WHILE in recent years there has been a gradually increasing rate of expansion of air transportation in most countries, the rate in Canada has been extraordinarily rapid and in fact since 1927 has been exceeded in only one other country.

As no Government subsidies have been paid in Canada, aviation has developed here along sound self-sustaining lines and each year discloses a greater increase in miles flown than the preceding year. Aircraft have been principally applied to uses and services for which they were particularly adapted and where they could do the work required more rapidly, cheaply or effectively than existing means. They have continued to be used only where they could compete with older methods of transportation.

Some three-quarters of the area of the Dominion are still unserved by roads or railways. Much of this area is rich in natural resources but has been little explored, owing to its relative inaccessibility by older methods of transportation. Employing aircraft, the exploration and development of these regions have been proceeding very rapidly during the last few years. Not only are aircraft used directly as a means of observation, but they are also employed for the transport of personnel and supplies of surveying and geological exploration parties and of construction crews, materials and machinery.

The airplane and aerial camera are proving invaluable aids to the surveyor, geologist and engineer for the location of railroads, highways and transmission lines, and in connection with river improvements, harbor, terminal and waterpower developments.

Valuable work is being done by aircraft in the protection and development of the forest wealth of the Dominion, in the detection and suppression of fires, combatting of insect pests by dusting and in enabling adequate plans to be arranged for the best utilization of the timber.

Experiments in the control of the wheat stem rust by dusting from aircraft indicate the effectiveness of this method of combatting a blight causing an average of $25,000,000 damage yearly. Fishery patrol by aircraft has also been demonstrated to be the most effective method yet devised.

In addition to services in connection with the national resources, aircraft are being employed in many other ways. The air mail is perhaps that best known to business men. Other applications include custom patrol, exhibition, sightseeing taxi, advertising and illustration and sport.

With the expansion of aviation in Canada, there is seen developing slowly but steadily the infant aircraft industry in support of and supported by the rapidly expanding flying services of the Dominion. The development and expansion of this industry are essential to the building up of an independent self-supporting Canadian aviation.—From a Canadian Section paper by J. H. Parkin, assistant director, division of physics, National Research Council of Canada.

Air-Transport-Design Economy and Air-Transport Testing

Aeronautic Meeting Paper

By Frank T. Courtney[1]

AIR transport is an essentially expensive business and operators must study economy at least as much as do other commercial companies to make their business self-supporting.

A commercial-transport airplane must have an even higher degree of efficiency than that of a large military plane and must be in continuous operation so far as possible.

Very high speed may prove to be an expensive luxury; it should be investigated with great care as well as with the present enthusiasm of its sponsors. Speed is not paramount, but profitable load-carrying is. A cheap plane that is inefficient and an efficient plane that is more expensive than it need be are both uneconomical.

Air-transport testing consists not merely in the elimination of defects but in the examination of all the characteristics of the plane to effect the maximum improvement that the design will stand and to discover other characteristics for future development or elimination. The author enlarges upon the foregoing and other economic and testing phases.

PASSENGER air-transport, after years of almost complete neglect, spread suddenly over the Country about three years ago. The basic idea in airline organization seemed to be that a newly air-minded public would pay whatever was necessary to travel by this new means. Airplanes were selected by the transport lines chiefly on account of their immediate availability in the required sizes and quantities. Such airplanes were being built mostly by people having little or no previous experience with large airplanes and the planes suffered from serious inefficiency, much of which was because they often were designed to suit some predetermined forms of construction rather than to satisfy calculated commercial requirements.

It was then found that the public would not pay the tariffs necessary to make the operation of such planes profitable but would only pay rates which the strictly economical operation of strictly economical planes would make possible. Then the history of air-transportation in Europe repeated itself in America, even to the merging of smaller lines into larger ones to satisfy Government-subsidy requirements. Operators in America now realize that they must regard the Government subsidy, in the form of mail contracts, as being of limited permanence and magnitude and must endeavor to get their business down to a self-supporting basis. Thus, in an essentially expensive business, they must study economy at least as much as do other commercial companies.

The first point that stands out in the producing of a commercial-transport plane is that it must have an even higher degree of efficiency than that of a large military plane. This point is worth emphasis. The commercial plane must be in continuous operation so far as possible; hence, say 100 lb. of avoidable weight or unnecessary loss in speed of a few miles per hour can count rapidly to become many ton-miles unprofitably carried or many hours of operating cost unnecessarily incurred. Yet such efficiency must not be obtained regardless of cost, as it may be in a military plane which is frequently a Government-laboratory job as well as a weapon. High efficiency, at low construction-and-operation cost, presents problems to the designer which are largely absent from military design. Emphatically, this paper is an attempt to study, not technical progress in design, but the economy demanded today, even though the two may prove identical.

A first and much-neglected step in economy is the study of history and profiting from the experience of others. One notices early in this study that the bright path of the "brain wave" rarely leads anywhere and usually turns back to the start; that the path to really successful design is along the tiresome and unromantic road of prolonged thought on a basis of hard facts and experience. There is a tendency, in the present stage of the industry, to try to get a jump ahead of the other fellow by new and "advanced" methods of design when full advantage has not yet been taken of known and tried methods and facts. What appear to be brilliant ideas of improvement are often introduced, at great expenditure of money, regardless of the fact that they are rather obvious and have therefore presumably been rejected in the past because they were found not to pay for themselves. The author speaks from some 18 years of flying experience in America and Europe, devoted mostly to testing, and following the history of, new machines and methods.

It is not suggested that new methods of obtaining performance or efficiency should be neglected, but there is a tendency to believe that older forms of design are incapable of providing the efficiency needed for the economy of today and that radically new departures are necessary. Yet a considerable improvement remains to be obtained from the development of such older types of design as fabric-covered parallel wings, externally braced with either wires or struts, in the monoplane, biplane or sesquiplane forms, in combination with con-

[1] Consulting test pilot, New York City.

ventionally built fuselages. Moreover, these have a background of years of research, experiment and knowledge, checked and rechecked. To advocate giving more attention to them at the present stage is not to advocate unprogressive or reactionary tendencies in design.

In aviation, new developments have always been extremely expensive in money and time. At a period when money is scarce and economical replacements are urgently needed, the cheapest and quickest line of procedure seems to be to apply to the refinements of older forms of design some of the ingenuity and thought now devoted often to wholly experimental new lines. At present, there is little or nothing to show that results will be inferior.

Performance

Speed is the chief factor in performance which is of interest to air transportation and to economy. It has been pointed out that the present uneconomical situation in American air transportation was caused by a false idea as to what the public would do. The author fears that an equally uneconomical situation is likely to arise from a false idea, increasingly prevalent, of what speed will do. At any rate, he feels that the attractive picture presented by very high speeds may prove an expensive luxury, that it should be investigated with great care as well as with the present enthusiasm, and that its sponsors should leave a way out in case of necessity.

Profitable Load Carrying Is Paramount

It frequently is stated that the first object of air transportation is speed. This is wrong; the first object is the profitable carrying of loads from one point to another, as in any other form of transportation. Speed is the second object. There has recently come into existence what may be called a super-speed school of thought which endeavors to prove that greater economy is effected by higher speeds. Such figures as the author has seen to that effect are usually too biased to be convincing. The use of very high speeds brings in a number of definite detail advantages; but, basically, aviation resembles all other forms of transportation in that costs go up rapidly and loads come down rapidly with every increase of speed; and loads decrease rapidly in proportion to time saved. Speed costs money; loads pay money. To justify itself, aviation must have a certain minimum speed advantage over surface transportation. If we have to pay for going above some such speed, we may be going out of economy into luxury.

The argument for high speed that usually is advanced is that the very high speeds permit usage of new types of design and result in new forms of efficiency. Within the speeds with which we are likely to be concerned, there are few, if any, engineering grounds to support this contention. One notices, to start with, that the world's fastest planes are merely refinements of very old forms of design. For example, there is great talk of the suppression of parasite drag, as if this were a new idea requiring new lines of attack; but it is not. The suppression of drag has always been one of the major objects of design, but it always has had to compromise with considerations of weight and maintenance. The ideas of retractable landing-gears and elaborate streamlining are neither original nor clever; it may be true that they contribute more in speed to the fast plane than to the slower plane, but it is equally true that their weight is much greater, in proportion to the payload, on the faster than on the slower plane.

Taking 125 m.p.h. as a desirable minimum cruising speed and comparing this with the 160 to 200-m.p.h. speed frequently discussed, it is seen that speed-increasing factors can be profitably used equally well in either case. The 125-m.p.h. plane can utilize the maximum permissible landing speeds; so the faster ship can gain nothing in that direction. For equally meritorious design, therefore, the faster ship obtains its speed almost entirely by the addition of power. The weight of this power, and of the fuel necessary to feed it, comes directly out of the payload. While there are admittedly modifying factors, this fact is a fairly close method of estimating what load must be sacrificed for speed, and therefore what that speed will cost.

In operation, the faster ship loses more in proportion by every delay, as in taxying, circling and approaching, and in the time taken to travel to and from airports. It is argued that the higher speeds will attract more revenue. Similarly, a higher load will permit a lower fare, which, from the economy viewpoint, is an even greater attraction, especially where the need is to attract the general traveling public rather than the air-minded enthusiast.

Economy demands the study, not of speed, but of the speed-load ratio. Doubtless, luxury speeds will be generally demanded and paid for in the future, but at present we are not out of the economy stage.

At any rate, the designer should assume that the speed question is still an open one. Economical design therefore would provide that the transport should be conceived so as to allow for a reasonable variety of uses without great engineering change in the initial design. If the initial design is such that this variation is not possible, any error in the forecast of requirements or any change in conditions will result in that design being like many transport planes today, too expensive to operate and too expensive to throw away. It is possible and even easy to arrange the initial design so as to allow for horsepowers, performances and payloads in various combinations; but this can be done only by first giving much thought to the design. It can rarely be satisfactorily done as an afterthought. By such study an economical type is more easily assured and, moreover, the constructor's range of prospective customers is increased without the necessity for a completely new design for each requirement.

Design Considerations

Operators seem to have come to the reasonable decision that a larger number of relatively small units, run on more frequent schedules, constitutes more economical equipment than do the "giant" types; the 10-passenger unit seems to be the generally accepted size. Complications may arise here, however. If one fixes this load and then fixes a suitable speed, it may be found that no particular engine is available for just this combination, and a somewhat higher power must then be used. How is this power to be applied? Shall the load remain fixed and a higher speed be used, or shall the speed remain fixed and provision be made for more load? One suggests that the latter alternative is the more economical, although the former offers the more superficial attraction in the form of speed and is, moreover, easier to work out.

Putting it in a simple form, if air-transport patrons continue to increase as they seem to be increasing today, there probably will be more operators who wish they had that extra couple of seats than those who wish they had that extra speed of 5 or 6 m.p.h. Again, as has been found frequently in the past, developments in operating requirements have resulted in the addition of all kinds of extra equipment or extra fuel, and the process is not likely to have ceased yet. Operators of planes with border-line payloads have often had to sacrifice much-needed payload to fit these needs. A surplus of load can be more usefully suited to varying requirements than can a surplus of speed. This is a point which is worth more than casual inspection. Again, it seems to the author that many economies in cabin space and

weight can be effected more easily in the 18 to 20-passenger size than in the smaller sizes, and on many lines, at the prospective rate of expansion of passenger traffic, such a size easily may be justified.

The number of engines to be used constitutes a problem. As a load carrier, the single-engined plane has definite advantages. This is due only to a small extent to any engineering or aerodynamic advantages inherent in the use of only one engine; it is chiefly because the multi-engined plane must limit its load to what it can carry after the failure of one engine, whereas the single-engined plane can load up regardless of such considerations. How long the single-engined plane will be held to justify that advantage is another subject. For similar reasons the three-engined plane has load-carrying advantages over those of the two-engined plane, these being largely offset, however, by a fairly large collection of minor factors; so, at present, the more economical type of plane, the three-engined, is likely to be gradually replaced by the two-engined plane.

Cabin Design Important

Cabin design offers much scope to the ingenious student of strict economy. Passenger comfort demands chiefly more leg room. To increase spacing between seats, however, involves a longer and heavier cabin and the passenger discomfort attendant upon being seated farther from the center of gravity in rough air. If we place three or more seats abreast, we reduce cabin length and weight, and the weight per passenger of roof, flooring and walls and their fittings. Increased body-diameter, however, increases drag. Headroom is an unnecessary luxury and offers scope for economy. Its few attractions do not pay for its drag or for the considerable weight resulting from long, unbraced members. The only excuse for passengers to move about in the cabin to any great extent is that they are too uncomfortable where they are sitting, and every pilot would bless a degree of comfort that would keep passengers in their seats.

In sorting out all these conflicting demands, one notices the great advantages to be obtained by the double-deck arrangement, even in the 16 to 18-passenger sizes. These advantages go beyond mere passenger distribution and allow for all kinds of variation of load of both passengers and freight with minimum disturbance of flying characteristics. The deep fuselage involved can be made to pay for itself further in reducing the weight of wing bracing. The point is raised merely as an illustration of directions in which economy may be sought.

Economic Factors Outlined

On the score of economy, the extended use of external streamline wire bracing is worth study. This is a typical case of an older form of construction which tends to be eliminated by "advanced" design but which has advantages that should not yet be neglected. Usually the elimination of the older form is based upon the suppression of parasite resistance; but, in fact, this argument usually is false. For high speeds, the thin wing with external bracing has less drag than the thick, internally braced wing, a fact which is made evident in all successful racing design. Again, the externally braced wing system is lighter than is the internally braced system. The advantages are so great that they continue to apply even to biplanes of large span where no attempt is made at tapering, which is an expensive necessity in the cantilever wing, and where many additional struts are introduced.

The author believes that the full advantages of external wire bracing have not yet been used and that economy demands at least further investigation. For example, the landing gear usually is a useless encumbrance except when on the ground. Therefore it is now frequently made retractable, at much additional weight and cost, to get rid at least of its resistance. If, however, the landing gear, apart from the wheels, is used as a part of the general bracing of the structure, as it is in many racing planes, it seems possible that the resulting gain in structural economy would lead to at least as great over-all efficiency as the retracting gear would give, with an additional saving in maintenance. Again, advantages obtained with small planes in this direction are multiplied in the case of large planes, where weight considerations present more problems.

Certain objections are raised to the use of wire bracing. Admittedly it seems to be a less solid job than other forms and appears "old-fashioned," but this is no argument where economy is concerned. In the past, wires have suffered from breakage, due largely to the effects of vibration, but modern metallurgy and engineering should provide methods of manufacture and of fitting which would remove the risk of excessive vibration and its effects on the wires. Again, the great economy of wire bracing permits of effective duplication. There is no evidence that it requires more maintenance or attention than other forms of structure; on the contrary, necessary adjustments and replacements are cheap and easy. Ice formation certainly presents a problem; but, in this connection, the streamline wire should not suffer from handicaps from which many other parts of the structure are free. The greatest risk is breakage due to vibration following ice formation, but both the vibration and the breakage should be rendered negligible risks by adequate study.

A great attraction of the cantilever is that it permits the use of metal skin-covering. Whatever may be the future of metal wing-covering, the fact is that, at present, its weight is too great, in whatever way it is used, to pay for such advantages as it offers. Even these advantages are much in dispute; there is no definite evidence yet that it has any serious maintenance advantage over fabric. One feels that metal covering is a sounder engineering job, and that it will eventually be universally used; but one suggests that, at the present stage, it is not economical.

In bringing forward the foregoing points, the author realizes that he lays himself open to accusations of upholding obsolete systems of design and construction. In reply, he claims that, where technical developments looking to the future are concerned, he is at least as much interested in new and progressive methods as anyone else; but a wide experience with those subjects has made him into a naturalized Missourian, and he is driven to think that, where the question is one of making money for the present and immediate future, the lessons of the past are a safer guide than the hopes of the future.

Engines and Other Features

The subject of engines is a large one in which fuel plays a large part. One might note, however, that the increasing use of supercharged engines, previously more or less confined to military planes, can give economical operation together with a reserve for emergency which helps the multi-engined design greatly. Another point of interest concerns the continued use of the radial engine. Its economy advantages are obvious, but what is not so obvious is the price paid for maintenance and for structural weight to withstand vibration. It may be found eventually that the saving effected by better inherent balance of in-line engines will justify further interest in them.

The position of the maintenance man, like that of the test pilot, usually is regarded as being one of importance only in the detail design. The author thinks that a really qualified maintenance expert, like a really qualified test pilot, should play his part in the general design, in

cooperation with the engineering staff. The position of engines, landing gears and the like and many other maintenance considerations that can greatly influence the aerodynamic and general design, are matters that must not be arbitrarily demanded by the operator or introduced by the designer. Too often today the designer will excuse some feature which impairs performance by saying "that's what the operator asked for," or the maintenance man will decry some feature which the designer included for some definite reason unknown to the maintenance man.

Finally, it is frequently argued that, since efficiency is so rapidly paid for in operation, first cost is unimportant in a transport plane. This does not, however, excuse expensive design and construction. No evidence is available to show that the expensive designs often indulged in today cannot be equalled and beaten by relatively cheap refinements of older forms of design and improvements in conventional construction. A cheap plane that is inefficient and an efficient plane that is more expensive than it need be are both uneconomical.

Air-Transport Testing

Probably no aspect of the producing of a commercial airplane is so neglected and even ignored as is the importance of its testing. In fact, it is almost fair to say that some of the transport types commonly in use have never been adequately tested. These may seem to be strong statements, yet it is possible to show that several fatal accidents have resulted from features which competent testing would have removed.

Testing usually consists of having a skilled pilot fly a newly completed airplane so that he can note the more obvious defects. Any trouble that does not prevent the fulfilment of minimum requirements is not considered as a defect, and therefore is not to be changed. As the business of the Department of Commerce is not to perfect the airplane for the constructor but to compel fulfilment of certain minimum conditions, many transport planes are permitted to fulfil these minimum requirements and little more. Whatever this process may be called, it certainly is not "testing."

Whether from an engineering or a commercial viewpoint, testing is something far more than the mere flying of a new plane by a skilled pilot. It consists, not merely in the elimination of defects, but in the examination of all the characteristics of the plane to effect the maximum improvement that the design will stand and to discover other characteristics for future development or elimination. This applies more to the transport plane than to the military or privately owned plane, for the following reasons:

(1) A transport plane must be expected to fly for many more hours than does any other type.

(2) Its controls should be correspondingly more easy and its general flying characteristics more nearly perfect.

(3) It must be expected to face more and longer periods of bad-weather flying, which demands a further perfection of all arrangements affecting piloting.

The foregoing requirements are further complicated by the comfort demands of passengers, the unusual variations of load and trim resulting from the carrying of passengers and freight, and by the requirements of maintenance, reliability, insurance and similar factors that are more important in the transport plane than in any other type. "Good enough" are words which every producer of a transport plane should forbid. Characteristics which are merely "good enough" are not good enough.

The factors leading to poor or inadequate testing are so usual as to be easy to point out. The testing generally is regarded merely as the final stage in the engineering of the plane, and comes directly under the engineering department. The pilot comes directly under the chief engineer and has the job of finding out what is wrong with the plane; in other words, of discovering what the chief engineer has done wrong. The chief engineer, being human, will make the minimum admissions of defects or of necessity for changes; moreover, he has other difficulties with which to contend. He frequently is working under non-technical executives who will regard the exposure of any defect or a demand for any considerable change as a reflection on his personal competence. What incentive has he, therefore, to employ a pilot who insists on delving deeply into all the characteristics of the plane and on recommending changes to things that are "good enough"? So, how can the pilot risk the antagonism of the chief engineer and the loss of his job by demanding anything more than minimum changes? In other words, how can the plane be really and thoroughly tested?

A single example will suffice to illustrate results. A multi-engined plane is produced and must be able to fly with one outboard engine cut out. It is taken up to a suitable height and one engine is stopped. If the pilot cannot, by any means, hold the plane straight, then there is a defect that must be corrected. But if, by the exercise of his complete attention and entire physical efforts, the pilot *can* keep the plane straight, and possibly turn slowly against the other engine, then the plane *can* fly with one engine cut out and it is considered "good enough."

The plane then goes out into commercial service, and some day an engine fails on the take-off. The transport pilot, flying with a load of passengers and under great responsibility as well as having other things to think of and deal with besides merely keeping the plane straight, finds that he cannot adequately control the plane and puts it down as best he can. Such cases have occurred, and thus resulting crashes are directly attributable to inadequate testing. It is possible, and even easy, to design rudder controls so that, on the failure of one engine, no particular effort is required on the part of the pilot to maneuver the plane, yet this is an exceedingly rare quality in transport planes. Failure to arrive at this result indicates definitely the absence of a real testing organization. Numerous similar examples could be given to the same effect.

Executive Supervision Desirable

The testing of the plane is the business of a separate and distinct part of the company organization. It should no more come under the engineering department than an inspector should come under an engineer whose work he has to inspect. An exception might be made for purely engineering tests, but these remarks refer to the commercial aspects of testing. Thus, testing starts with the executives, to whom the test pilot should be directly responsible. Just as they must select a competent chief engineer on the basis of his qualification for the job, so they must select a test pilot.

Mere skill in piloting is a minor qualification for a test pilot. Frequently it is the only qualification demanded. But the pilot must have a through understanding of the aerodynamic and engineering problems of the engineer. This does not demand that he shall be equally competent to solve them; but, unless he understands them, he cannot make intelligent attempts at constructive criticism. Above all he must have the widest possible experience with different types of plane, their experimental histories of success or failure and their good and bad details. Without this knowledge it is impossible for him to know whether any feature in the current job is or is not capable of useful improvement.

It is, in fact, on the prospective test pilot's record in this respect that the executive will know how to select him. Instead, he is often selected purely on his reputation as a skillful pilot; or on his having performed some spectacular feat; or, worst of all, on his being particularly familiar with the company's former products, thus confining him to a narrow outlook.

As regards transport planes, a test pilot must have, in addition, experience concerning the work and necessities of the transport pilot, for, without that, he will leave undone those apparently small but important improvements that make all the difference to transport flying. The work of the really experienced test pilot does not commence merely when the plane is ready to fly. In the original stages of the design, much money and time can be saved by the test pilot if his criticisms of features which he knows, from experience, will prove unsatisfactory on test are heeded.

In addition to the correct selection of a test pilot, the executive has the additional responsibility of seeing that the set-up is such that the pilot, and the engineering staff of which he is the professional critic, shall work in harmony. This is a natural difficulty in testing, yet it is the job, not of the pilot or the engineer, but of the executive. This is possible only where both engineer and pilot are experienced men; where it is achieved, highly satisfactory results are likely to be obtained.

Unbiased Testing Is Essential

Outside experience and advice in testing are rarely effectively used, are often resented and yet can be very valuable, just as in other professions. A detached and unbiased attitude toward the product can lead to many discoveries that might be missed by those who have watched their own baby grow up. Moreover, a pilot whose experience is confined to the work of one company is likely to miss the relative merits of features developed outside. It is often found that an experienced test pilot welcomes the opinion of another experienced man, but the executive feels that, if he is paying one good man, there is no point in paying another. The inexperienced test pilot always, and naturally, resents the interference of an outsider. However, outside help is sometimes sought by executives in the worst possible way. Often, when the plane is completed and has been "test-flown", several "famous pilots", after an excellent lunch, are permitted to fly the plane. Politeness being their main obligation, a "yes-man's" chorus of praise results, from which nothing but harm can come, especially to the real test pilot who has been endeavoring to obtain improvements in what now turns out to be the perfect airplane.

One would suggest, further, that the manufacturer should not seek his test information directly from the airline pilots. This does not mean that he should ignore subsequent recommendations from them. It does mean that it is a practical impossibility to attempt to suit the varying opinions in the first instance from a number of men whose experience is outside the range of testing. It is the duty of a competent test pilot to seek and to analyze the opinions of airline pilots, and to see that the generality of opinions is introduced by him in his test work. Where that is done it is highly unlikely that the airline pilots will be otherwise than completely satisfied, and inefficiency from diversified responsibility is avoided.

Increasing competition in the transport field necessitates the highest quality in the product, which is very much more than the present demand for merely scraping past the requirements of the Department of Commerce. Any Department inspector will agree that, where the manufacturer tries to go well beyond the demands of the Department, it is all to the good of the Department, the particular manufacturer and the industry in general. It demands nothing more than the careful organization, on the part of the executives, of the whole system of the testing of their planes, in place of the prevalent system of merely flying them.

The author believes that no plane can be regarded as having had its controls properly tested until the test pilot has done a considerable amount of blind flying with it. Blind flying is the most effective and quickest way of showing up imperfections. If some of the transport planes now in use had been flown blind for 15 min. with one engine cut out, the author is reasonably certain they would never have been put into service as they are.

Landing gears need much comment from test pilots. Many of these are devised so that they never use, in practice, the travel of which they are theoretically capable. This results in harsh landings, discomfort to passengers and strain on the structure. Some planes are so very good in this respect that it seems unnatural and uncommercial that these qualities are not universally obtained. Under present conditions of testing, however, any comment from the test pilot in this connection is likely to lead to his being told to go and learn how to land smoothly.

Points That Should Be Examined

It may be of interest to refer to a few examples of points that the test pilot of a transport plane can profitably examine. There is a clear, practical dividing line between stability and instability, in any practical sense clear enough, at any rate, to allow the various stability combinations to be worked out. Positive stability is, however, a matter of degree. Insufficient stability will mean excessive flying work; excessive stability will mean excessive response to disturbing air-currents and consequent passenger discomfort. Laterally, this usually is simple to deal with, though often a variation in dihedral must be met with some change in fin area. Longitudinally, stability can as well be obtained with a short-coupled stabilizer of large area as with a longer tail of less area; but, in the former case, trim variations are harder for the pilot to deal with, and pitching in rough air is worse. Therefore, the test pilot is called upon for comment beyond merely seeing whether there is stability or not. This is an example of where the test pilot's advice should be acquired at the start of the design.

Directional stability in multi-engined planes is considerably complicated by requirements of flight with one engine cut out. In the author's experience this largely involves study of the relative quantities of fixed and movable vertical areas. Rarely, except by luck, is this right the first time, and it is a point that repays considerable study.

Few transport planes exist in which the actual controls could not be improved considerably. That is a typical case of passing what is "good enough." In many cases, where the controls are quite good, much improvement can still be effected by some variation of gearing. The commonly used Friese-type aileron balance is capable of many variations. One often finds it overbalanced at small angles, leading to a disinclination to return to center, with a consequent impression of lateral instability. It is possible to remedy this without reducing the balance at larger angles. The qualities of the elevator control are complicated by the possible variation of load distribution in the cabin while in flight and by the effect of the engine thrust-line. The smallest possible elevators are desirable for the pilot's comfort, but where passengers can move about over a long cabin, or where variations of throttle setting greatly affect trim, more powerful elevators, with consequent attention to

(*Concluded on p. 377*)

Indicators as a Means of Improving Aircraft-Engine Performance

Aeronautic Meeting Paper

By Ford L. Prescott[1]

OPTICAL, balanced-pressure, electrical and sampling-valve indicators are discussed with particular reference to their shortcomings.

Sampling-valve indicators possess the advantage of simplicity, as compared with the balanced-pressure type, and make a permanent record on ordinary indicator paper.

The sampling or averaging type and the electric or instantaneous type each has fields of usefulness not covered by the other. The electric indicator is claimed to be the most satisfactory device for transient and qualitative work such as combustion study, but the sampling type is said to be better suited for engine development, valve-timing studies, supercharging and similar work.

STUDY of engine-pressure indicators was undertaken by the Materiel Division of the Air Corps some two years ago, and a comprehensive study of available indicators, together with development of the most promising types disclosed by the tests, has been carried out. The obvious purpose of the tests was to procure or develop satisfactory equipment for the study of pressure phenomena in high-speed, high-performance aircraft-engine cylinders.

Preliminary study led to discarding, as unsuited to the purpose in view, the micro-indicators and the small high-speed indicators of the conventional type. The reasons for this were inertia of the pencil mechanism and of the recording drum and also the small size of records obtained. These indicators are not adapted to taking weak-spring cards, which is essential to a study of pumping loops.

Indicators of the optical type employing moving parts of appreciable mass, such as the Midgley, were discarded because of inertia effects and difficulty of installing on an engine. The Midgley indicator, however, has one obvious advantage over some previous optical types, in that the piston is located within the combustion space, preferably flush with the wall. Thus, any question of pressure lag or surge in a connecting passage is eliminated. A moving piston exposed to the hot gases eventually sticks by reason of accumulated carbon and thus becomes unreliable. This type of indicator does not, therefore, appear to be suited to research work. The jagged pressure record on detonation, once supposed to represent detonation waves, is believed to be due to the natural frequency of the indicator mechanism.

Balanced-Pressure Indicators

Progressing to the balanced-pressure type, we have the R.A.E. and the Bureau of Standards indicators. Both of these operate on the principle of balancing an externally controlled pressure against the instantaneous cylinder pressure and require an external source of pressure and vacuum exceeding those to be measured. At the instant of balance, an electrical contact is made or broken. This contact point in the R.A.E. indicator is a light valve, seating on either side and having a travel of between 0.005 and 0.010 in. from one seat to the other. Upon leaving either seat, contact is broken and the primary current of an induction coil is interrupted. A spark passes from the pressure-actuated stylus and punctures a hole in a paper on a drum revolving at engine or half-engine speed. By this means, a pressure-time diagram is obtained. The shape of the pressure peak varies from one cycle to another, and for this reason the high-pressure portion of the record obtained with the R.A.E. indicator is composed of a mass of widely scattered points, through which a line is drawn at one's own discretion. Eliminating this defect appears to be impossible, hence the record loses value. A measurable time is required for the disc valve to pass from one seat to the other. Time is also required to build up sufficient flux in the induction-coil core to cause a spark to pass, thus imposing a definite speed limit, beyond which only the rising side of the pressure record is obtained. With this indicator, low-scale or weak-spring cards can readily be obtained. Some lag due to pressure required to actuate the moving stylus undoubtedly occurs, which renders such low-pressure cards of little value unless taken very slowly.

The Bureau of Standards indicator is without doubt the most precise of those tested during development. A light steel diaphragm is supported between two perforated discs and touches an insulated electrode at approximately atmospheric pressure. A timer is driven at half-engine speed and closes contact during a brief instant, about 2 deg. of crankshaft rotation. If, during this instant, the cylinder pressure is higher than the external pressure, contact is closed and a click is heard in a telephone receiver. If the external pressure is higher, no click is heard. At the point where clicks cease, the external pressure equals the cylinder pressure during the short period of contact. Thus, a series of points is obtained and plotted against crank angle, or piston position, giving a pressure-time or pressure-volume card. During the high-pressure portion of the card, estimating approximately 50 per cent clicks is necessary for a mean value or 0 and 100 per cent clicks must be secured for maximum and minimum pressures at the crank angle in question.

In working in the same room with an engine, the clicks could not be heard; therefore a circuit was devised, including an induction coil having a neon lamp in the secondary circuit. The flash could readily be ob-

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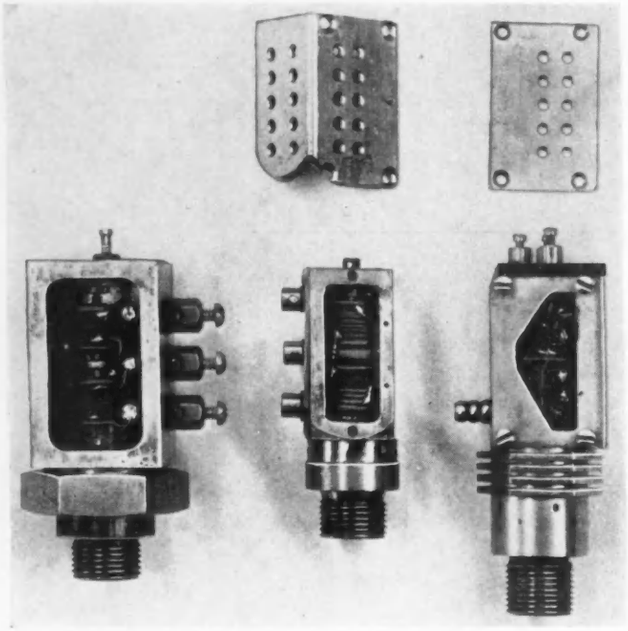


FIG. 1—TELEMETER OR CARBON-PILE PRESSURE ELEMENT

The Principle Underlying This Type Is the Variation in Conductivity with the Pressure Applied to the Pile of Carbon Discs. Two Stacks, Each Having a Resistance of Approximately 45 Ohms, Are Opposed and Connected to a Diaphragm So That Any Increase of Pressure on One Stack Is Accompanied by a Corresponding Decrease of Pressure on the Other

served and this enabled cards to be taken with the ears protected from the deafening exhaust of the engine. Two operators can take a card in about 10 min. by reading every 10 deg. of crankshaft rotation. This consumption of time and lack of permanent record, other than a table of observed angles and pressures, constitute a major objection to the indicator. For lower-loop diagrams, a shift of the zero position of the diaphragm with temperature occurs, which may amount to 1½ lb. per sq. in. and constitutes an error of 100 per cent if pressures are measured within this limit. Without introducing a cock and a considerable length of connecting pipe, the position of the atmospheric line is uncertain. Hence, on lower-loop diagrams, some doubt exists as to where the zero line should be drawn, and the same objection applies to the boost line on supercharged engines. This error is insignificant on the high-pressure card, and very close checks have been obtained on single-cylinder engines between the indicated mean effective pressure obtained from the card and the sum of brake and friction mean effective pressures.

An attempt was made to combine the Bureau of Standards diaphragm element with the R.A.E. recording mechanism, by constructing a relay to open a primary circuit whether the diaphragm contact opened or closed. Thus, a single contact on the side of the diaphragm not exposed to the products of combustion would cause a spark on the make as well as the break of contact. This worked beautifully at low speed, but at the higher speeds the electromagnetic lag was so great that complete failure of the record occurred. Double contacts on the diaphragm were unsuccessful, due to the rapid accumulation of carbon and moisture on the under side of the diaphragm. Water-cooling of the diaphragm element at times caused condensation of moisture on the diaphragm and support, causing them to stick together, with failure to record. The chief objection to the Bureau of Standards indicator is the time required to take data and plot them. Besides great accuracy, it has the advantage of simple application to an engine, since only a tube and a wire are connected to the pressure element, and recording may be done at a remote point.

An incident that shows how easily a foolish blunder can be made occurred in connection with an unsuccessful attempt to indicate a Diesel engine with the R.A.E. instrument. Cylinders of carbon dioxide have long been used to furnish the pressure to balance the disc valve of this indicator. At ordinary room temperatures this gas has a pressure of about 800 lb. per sq. in. and, as maximum pressures of from 1000 to 1200 lb. per sq. in. were expected, the "happy" idea of the moment was to place the carbon-dioxide cylinder in a bath of warm water. This was carefully done, and a pressure of 1200 lb. per sq. in. obtained. Upon attempting to take a card, the warm vapor condensed in the piping and valves and, when the pressure was blown off, the entire system froze up with carbon-dioxide snow. No cards were obtained. On a previous occasion a bottle of oxygen was used with the R.A.E. indicator and a cylinder head was blown off. Nitrogen or helium should be ideally suited to the purpose, but neither of these gases was available. The necessity of an external supply of high-pressure gas constitutes an objection to the balanced-pressure type of indicator.

Electrical Indicators

Two types of indicator that produce an electric current varying with pressure have been very successful for certain types of indicator work. These are the carbon-pile, or telemeter, type, as developed by Martin and Caris, of the General Motors Research Corp., and the condenser-transmitter type, developed in Germany and reported in the 1930 yearbook of the D.V.L. Both of these indicators use an oscillograph as the recording mechanism.

The condenser-transmitter type utilizes the principle of semi-resonance between an oscillating circuit and a second oscillatory circuit connected to the grid of a tube biased for detection. The detector circuit is detuned slightly, so as to operate normally on the steep side of the resonance curve. The condenser transmitter shunts the tuning condenser of this circuit, and, since its capacity changes with pressure, the oscillatory circuit is tuned or detuned by the action of pressure on the transmitter. The plate current of the detector tube is passed through an oscillograph and a pressure record thus obtained. Formerly, the transmitter condenser was constructed so that pressure moved the plates closer together. This gave low sensitivity to pressures near atmospheric and increasing sensitivity with higher pressures, while a vacuum would produce the lowest sensitivity. In the D.V.L. indicator, a small diaphragm is exposed to the pressure, and a strut transmits movements from this to the condenser diaphragm, mounted on the side of an insulated plate remote from the pressure source. Thus, pressure tends to separate the condenser plates, giving a zero position such that the plates are normally very close together. High sensitivity therefore is obtained with pressures near atmospheric and decreasing sensitivity at high pressures, with no danger of short-circuiting the condenser by the plates touching.

The telemeter type makes use of the principle of the varying conductivity with pressure of a pile of carbon discs. To secure a straight-line response, two stacks are opposed and connected to a diaphragm so that any increase of pressure on one stack accompanies a decrease of pressure on the other. The stacks have resistances of approximately 45 ohms each and are connected as two legs of a Wheatstone bridge. The galvanometer current of the bridge circuit is passed through a sensitive

oscillograph. Fig. 1 illustrates three of these pressure elements with cover plates removed to show the carbon stacks. A three-wire cable and a tube to conduct cooling air are the only connections required, and the diaphragm is mounted preferably flush with the inner surface of the combustion chamber. For recording, a special oscillograph, illustrated in Fig. 2, is required, because of the small current available from the pressure element. The oscillograph incorporates, as the other two legs of the Wheatstone bridge, two fixed resistors of 40 ohms which are connected to the ends of a 10-ohm potentiometer, the center movable contact furnishing, with the center tap of the two carbon piles, the two galvanometer connections of the bridge. These are connected directly to the terminals of the permanent-magnet oscillograph element employed to record the pressure changes. A rheostat and an ammeter are employed to adjust the bridge current, and a milliammeter, which can be cut in or out of the galvanometer circuit, is included for calibration and zero adjustment.

Optical-Type Indicators

The optical system employs a carbon-arc lamp with its condensing lens, a vertical-slit shutter of the cylindrical type, an auxiliary slit to trim the haze from the sides of the light beam, a permanent-magnet oscillograph element, a reflecting mirror to double the optical path, a cylindric condensing lens and a rotating or oscillating mirror to reflect the light to its focus on the curved screen. A shield is provided so that the image can be viewed in daylight and a camera built in so that it can be slid into the place occupied by the screen and the record photographed. When using the pressure-volume mechanism, shown in the upper view of Fig. 3, the light spot is continuously in sight on the screen and every cycle can be observed. The lower view shows the rotating-mirror assembly for taking pressure-time records. The mirror is of fused quartz with four optically plane surfaces accurately located at right angles to each other and parallel to the axis of rotation. It is driven at 1/20 crankshaft speed by a 20:1 worm reduction. Every fifth revolution is thus shown on the screen, and two consecutive faces of the mirror reproduce one complete cycle. The length of screen or film is a little longer than one revolution of the crankshaft as recorded on the film. These two mirror units are interchangeable without affecting the focus of the cylindric condensing lens, since the mirror surfaces at mid-position have the same relation to the lens and screen.

Fig. 4 shows the operating mechanism of the oscillating-mirror assembly. A small crankshaft is fitted with a miniature connecting-rod having an L/R ratio of 4:1. When other L/R ratios are used, a small error in phase occurs; for example, with $L/R = 3.2$, the maximum error in position of light beam is 1.6 per cent. Since the curves are relatively flat near the center and the error becomes zero at either extreme, the error in area of card is probably less than 0.8 per cent. The mirror is oscillated by a crank on the end of the mirror shaft, the free end of which mounts a lever having two equal arms of the same length as the crank. The lower end of the lever is mounted on a horizontal vibrating link, and the upper end has true straight-line motion and thus is adapted to guide the small end of the connecting-rod. The maximum error in the angular displacement of the mirror, referred to the linear motion of the straight-line mechanism, is about 0.6 per cent. Hence, for all practical purposes, the inherent errors in the mechanism are less than other experimental errors. The width of line drawn by the moving light spot is in excess of 1/16 in. Hence, a larger error is possible in interpretation of the card.

The special oscillograph as described has an optical arm of about 35 in., and a high-sensitivity galvanometer is used, giving a deflection of about 0.6 in. per mil. amp. Thus a bridge current from a 6 to 12-volt source, of only 100 to 400 mil. amp., gives cards of satisfactory size. The film used is $3\frac{1}{4} \times 5\frac{1}{2}$ in., and, with maximum sensitivity, the lower, or pumping, loop can readily be studied. The scale under these conditions is probably about 50 lb. per in.

The instrument is not a precision indicator, by reason,

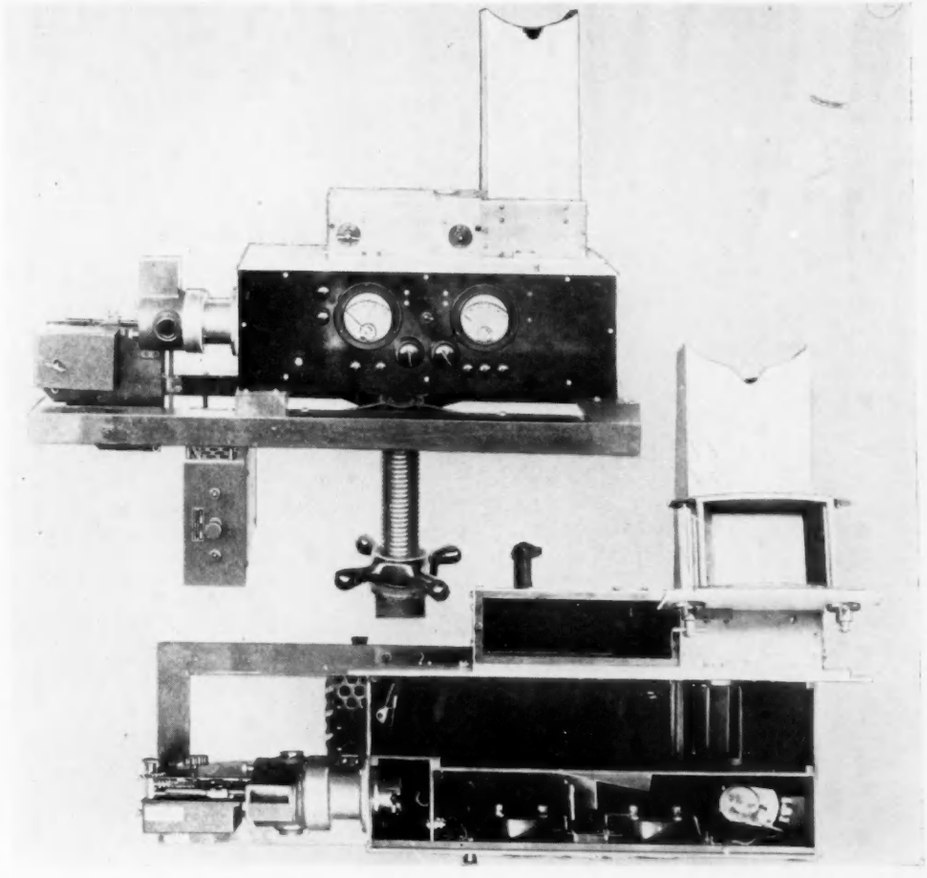


FIG. 2—TWO VIEWS OF A SPECIAL ELECTRIC-INDICATOR OSCILLOGRAPH

This Instrument Is Used in Conjunction with the Telemeter Shown in Fig. 1 To Obtain a Record of the Pressure Changes as the Current Available from the Pressure Element Is Very Small. The Smaller View Shows the Front of the Instrument, While the Other Is a Top View

first, of the varying conductivity of carbon with temperature. The electrical characteristics are also altered by the intensity of pressure, and the dynamic and the static calibrations do not check. Compressed air must be blown through the pressure element to cool the diaphragm and carbon stacks. The pressure element is affected by vibration and shock, and when inserted in the conventional aircraft-engine cylinder, a very ragged line is drawn because of the shocks transmitted to the carbon stacks (See Fig. 5). The frequency of this vibration, as recorded, appears to be that of the steel tongue which transmits pressure to the stacks. When used in a cast-iron cylinder, the records are free from this defect. The electrical connections are fragile and do not last long where vibration is severe. Obtaining a zero line under the actual test conditions is impossible.

However, in some problems this type of indicator offers the only available means of solution. It does produce a record of one continuous pressure wave, either recurrent or transient, and shows all the variations that occur, since each cycle can be observed. It has been used to good advantage to record the pressure variations in an engine oiling system, due to register of oil grooves and teeth in the gear pump (See Fig. 6). Performance of fuel-injection nozzles has also been studied and the effect of length and diameter of tubing for fuel injection have been determined from the records reproduced. Detonation pressures are recorded (See Fig. 5) showing the very high, if not actually instantaneous, rate of pressure rise when detonation occurs. Even with a very steadily operating single-cylinder test engine, it has shown a great variation, from cycle to cycle, in the time of maximum pressure or the time at which detonation occurs. All things considered, this indicator, while useful for certain special problems, cannot be considered satisfactory for ordinary engine research. The cost is prohibitive, as the special oscillograph required for satisfactory operation is necessarily expensive and not well adapted for general use as an oscillograph.

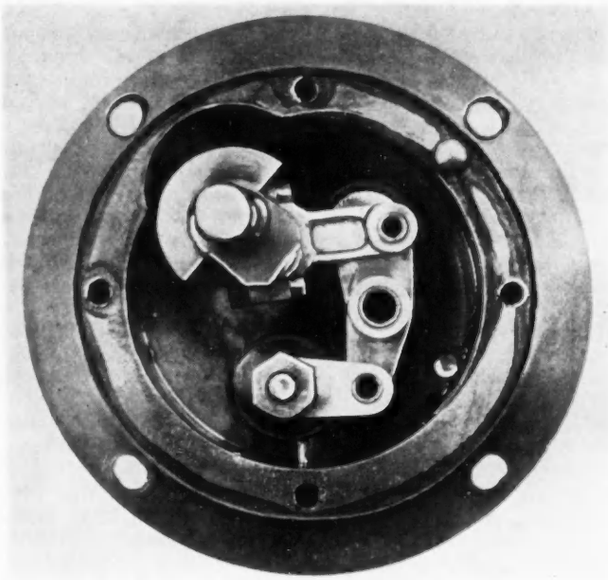


FIG. 4—OPERATING MECHANISM OF THE OSCILLATING-MIRROR ASSEMBLY

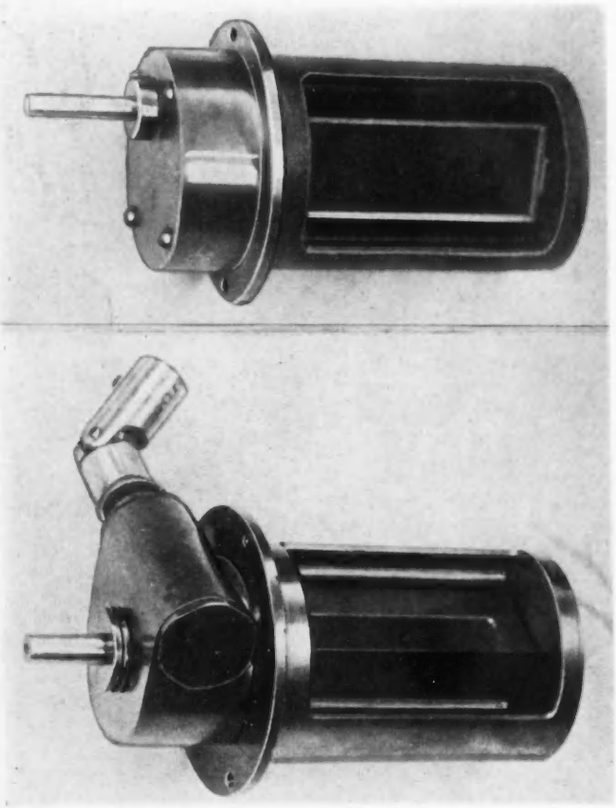


FIG. 3—TWO MIRROR ASSEMBLIES USED WITH THE OSCILLOGRAPH

That at the Top Is the Oscillating Type for Obtaining Pressure-Volume Records and the Lower One Is the Rotating Type for Pressure-Time Records. Both Units Are Interchangeable

Much the same criticisms would apply to the condenser-transmitter type of indicator previously mentioned. Some of the records reproduced in the D.V.L. yearbook show the effect of vibration just as in the case of the carbon-stack indicator. In addition to an oscillograph as required for the latter, the condenser-transmitter type requires the electrical apparatus incidental to the tuned semi-resonant circuits, such as vacuum tubes; variable condensers; A, B and C-batteries and meters for adjusting the circuits. These introduce new variables and increase the difficulty of calibration. The conclusion thus far reached regarding the electrical type of indicator is that for qualitative work, such as combustion research, development of fuel-injection systems and allied special problems, it is the best device available. The exact magnitude of the pressures as recorded cannot be relied upon, but the character of the pressure phenomenon probably is correctly recorded.

Sampling-Valve Indicators

One more method of indicating high-speed engines remains to be considered. This is the so-called "sampling," or point-by-point, method and has much to recommend it. A valve is interposed between the combustion space and an ordinary low-speed indicator and is driven at camshaft speed by a phase gear which is constructed so that the phase, or point in the cycle at which the valve opens for a brief instant, is manually adjustable. As this period of opening is moved through the cycle, the drum of the low-speed indicator is moved a proportionate distance. Continuous records are thus made of points taken from 1000 or more cycles of the engine. The phase gear can be driven automatically at a low speed or operated manually by a crank. The latter is prefer-

able, since the rate of change of phase should be proportional to the rate of change of pressure. Therefore the indicator should be moved very slowly where the pressure is changing rapidly and can be moved rapidly on the flat portions of the curve. Greater accuracy results, together with a saving in time. A complete card can be taken in 2 or 3 min., including change of indicator springs to secure both high-pressure and low-scale cards.

The best commercial example of this type is probably the De Juhasz, which is illustrated in Fig. 7. The valve of this indicator consists of a small piston surrounded by a sleeve working in an air-cooled cylinder. A port through the piston registers with a port through the sleeve twice each revolution of the indicator crankshaft. At the same instant a port through the cylinder registers to give a free port area through the entire unit. The action of the port is rapid, due to the piston and sleeve moving in opposite directions, even though the indicator is driven at one-fourth engine speed. A tube, about $\frac{1}{8}$ in. inside diameter and varying in length from approximately 4 in. as the minimum to approximately 12 in. as the maximum, connects the valve with the combustion space. The tube requires water-cooling, either with a wet rag or by soldering a second tube to it and flowing water through the latter.

Tube-resonance effects might be thought to produce a very ragged card, but in practice any pressure surges that may be present are lost in the average card, due to the fact that they have no definite time relation to the cycle. Shifting of pressure peaks due to irregularities in combustion is much more serious and can be taken care of by throttling the indicator cock, thus making the card conform to the average cycle, instead of attempting to follow the individual cycles. The time lag due to a connecting tube 11 in. long is approximately 0.001 sec., based on the transmission of pressure at the velocity of sound. Cards have been taken with this indicator, using tubes longer than 12 in., at an engine speed of 1700 r.p.m. and also with varying lengths of connecting tube, but little or no difference is observable in the cards until a length of 12 in. is exceeded. If the engine is operating at 1800 r.p.m., or 30 r.p.s., the delay should be $(30/1000) \times 360 = 10.8$ deg. of crank travel. That this delay does occur is entirely possible, but the pressure possibly may be transmitted at a velocity greater than 1100 ft. per sec.

This indicator uses a standard outside-spring Maihak indicator for the recording mechanism, and the phase gear provides a crank and slider mechanism having an adjustable L/R ratio with which pressure-volume cards are taken. Offset, or 90 deg., cards provide a means of

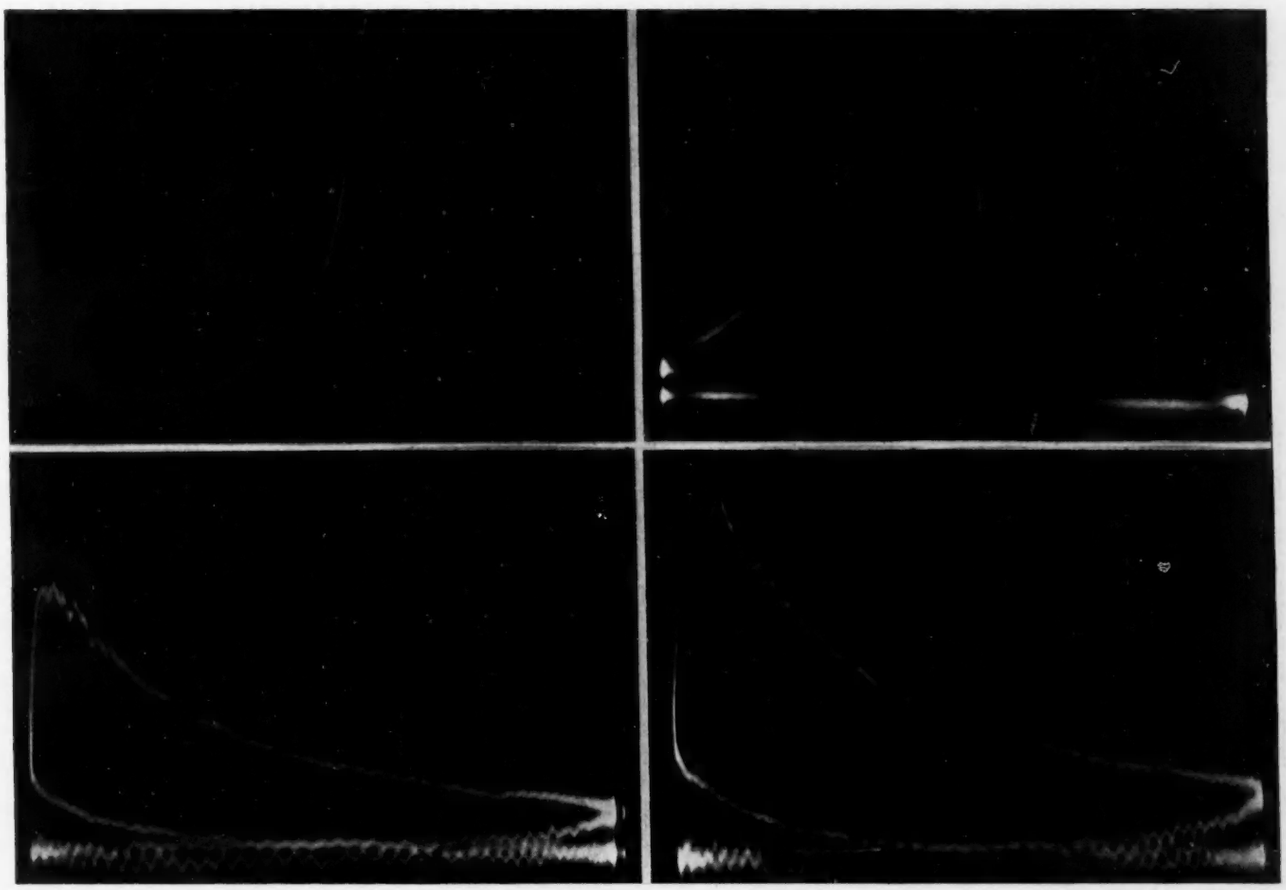


FIG. 5—EXAMPLES OF CARDS OBTAINED WITH THE CARBON-PILE INDICATOR AND THE OSCILLOGRAPH FROM AIRCRAFT ENGINES

(Upper Left)—Pressure-Time Card with a Steel Cylinder. (Upper Right)—90-Deg. Offset Pressure-Volume Card with a Cast-Iron Cylinder Showing a Detonation Peak with a Great Variation in the Shape and Time of the Pressure Peak. (Lower Left and Right)—Typical Pressure-Volume Cards with a Steel Cylinder. In All of the Cards, Except That with the Cast-Iron Cylinder, the Effects of Vibration of the Pressure Element, Which Are So Bad as To Render the Cards of Little Value, Are Apparent. In the Case of the Cast-Iron Cylinder, the Poor Record Is the Result of Using an Alternating-Current Arc, Instead of Direct Current

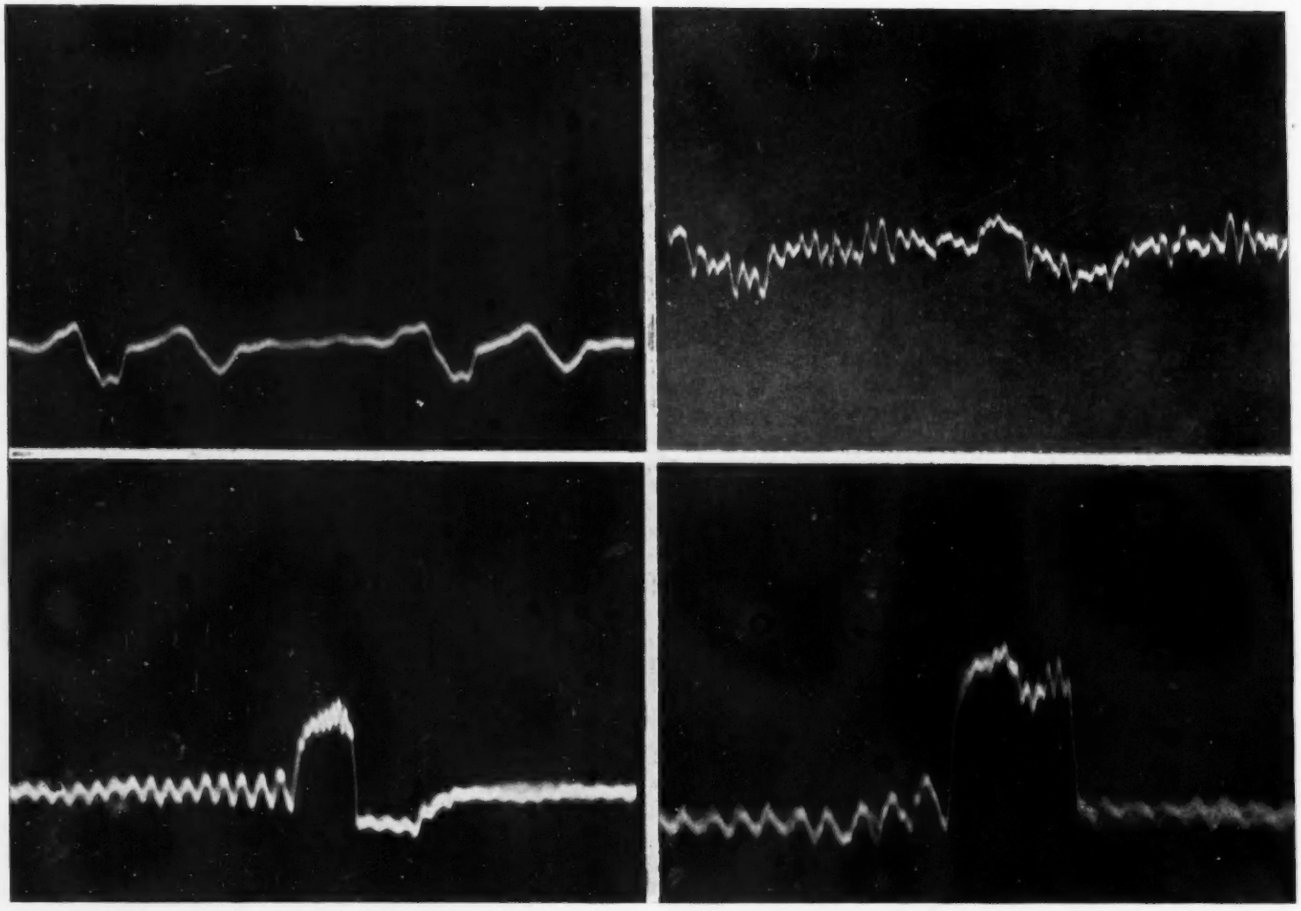
studying the peak of the pressure diagram substantially in pressure-time form.

Maintaining the piston valve tight enough for the correct register of the high-pressure portion of the card is very difficult. The port area is large, and the indicator stylus attempts to follow the variations of pressure from cycle to cycle, hence the pressure peak is very ragged. Pressures up to 150 lb. per sq. in. are held without difficulty, but higher pressures blow the oil seal from the piston valve and the record appears to be consistently low. In one case, where the pressure peak was known to have a value of approximately 475 lb. per sq. in., the record showed only 350 lb. Results were somewhat better when castor oil was flooded over the valve just before taking the high-pressure portion of the card.

The use of a connecting tube appears to be a necessity, and this entails an unknown phase shift, as previously pointed out, which may be objectionable. The phase gear and valve are in one unit. Hence, some installations are not as convenient as would be the case were the valve a separate unit. However, the weak-spring, or lower-loop, diagrams are excellent and for valve-timing and inlet-manifold development are most useful.

The multiple-element sampling valve or direct-pressure indicator of Commercial Laboratories, shown in Fig. 8, is another interesting example of this type. It is made with as many as 12 pressure-sampling valves, located at the wall of the combustion chamber, each actuated by an electromagnet and timed so as to operate in the firing order of the engine. Simultaneous pressure records are made in phase on a wide sheet of paper. The phase gear and electrical timer are driven by a flexible cable from the engine crankshaft. The phase gear can be engaged or disengaged, and the valve electromagnets can also be cut in or out. A special storage-battery is used to actuate the valves, and, on account of the heavy drain, the valves cannot well be operated simultaneously. A circuit has been devised whereby the phasing can be checked with a neon lamp. The mechanical drive of the phase gear is open to some objection, since, if slow enough for the pressure peaks, it is too slow for the rest of the card. Flexible shafts must be looked upon with suspicion, and the magnetic lag of an electromagnet is proportional to the speed. The latter is compensated for by increasing the voltage applied to the electromagnets. The pressure recorders are inverted and filled with heavy oil above the pistons and are of the outside-spring type. Springs can be changed to secure weak-spring cards if desired. The most obvious use of such an indicator is to study distribution, manifolds and ignition, not attempting to analyze the cards. For this work, the indicator has given good results.

Recognizing the good points of this method of indi-



Card Taken at the Oil-Pressure-Gage Connection of a 12-Cylinder Aircraft Engine Showing the Effect of Oil-Groove Register at an Engine Speed of 600 R.P.M.

Pressure Cycle of a Fuel-Injection System at High Speed

Another Card Taken at the Same Point Showing the Effect of Oil-Groove Register to a Slight Extent and also What Appears To Be Pulsations Due to the Teeth of the Gear-Type Oil-Pump

Pressure Cycle of a Fuel-Injection System at Low Speed

FIG. 6—OTHER EXAMPLES OF PRESSURE-TIME CARDS FROM AIRCRAFT ENGINES

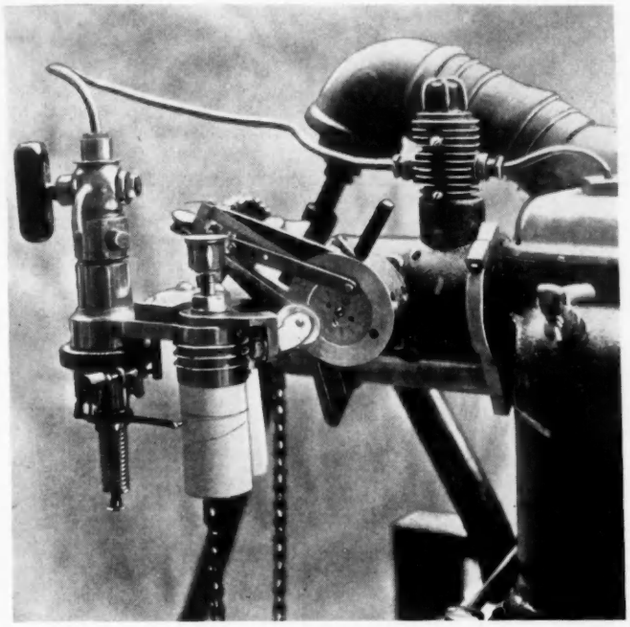


FIG. 7—DE JUHASZ SAMPLING-VALVE INDICATOR APPLIED TO A KNIGHT ENGINE

cating high-speed engines, the Materiel Division has developed a highly satisfactory indicator, shown in Fig. 9, which has given excellent service in engine-development work. The requirements laid down for this indicator were as follows:

(1) Complete elimination of connecting passage by locating the valve in the combustion-chamber wall

(2) Positive mechanical drive

(3) Minimum duration of valve-opening period

(4) Elimination of leaks in the valve and recording system

(5) Minimum backlash in the phase-gear mechanism

These requirements were most fully met by a poppet valve in a unit that can be screwed directly into the combustion-chamber wall. This unit is driven through a light shaft and universal-joints, by a simple phase-adjusting gear which, in turn, is mechanically driven from the engine or dynamometer shaft. A special form of balanced valve relieves the valve spring of load due to pressure trapped in the recording system. Gas-tight packing is used, with pressure lubrication from the engine oiling system, making the valve remain tight for long periods. As a further precaution, a cooling passage is provided for water circulation in the valve body.

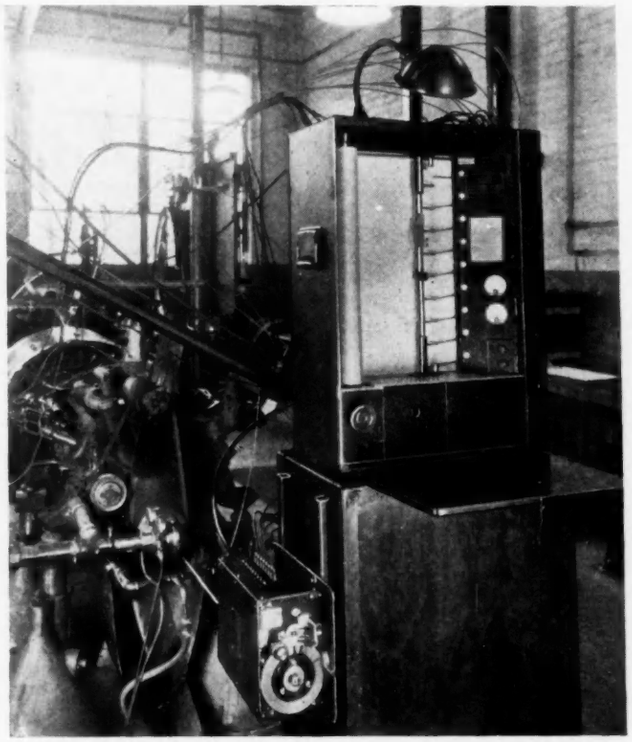


FIG. 8—A MULTIPLE-ELEMENT SAMPLING-VALVE INDICATOR

This Indicator Can Be Supplied with as Many as 12 Pressure-Sampling Valves, Each Actuated by an Electromagnet and Timed so as To Operate in the Firing Order of the Engine. Simultaneous Pressure Records Are Made in Phase on a Wide Sheet of Paper

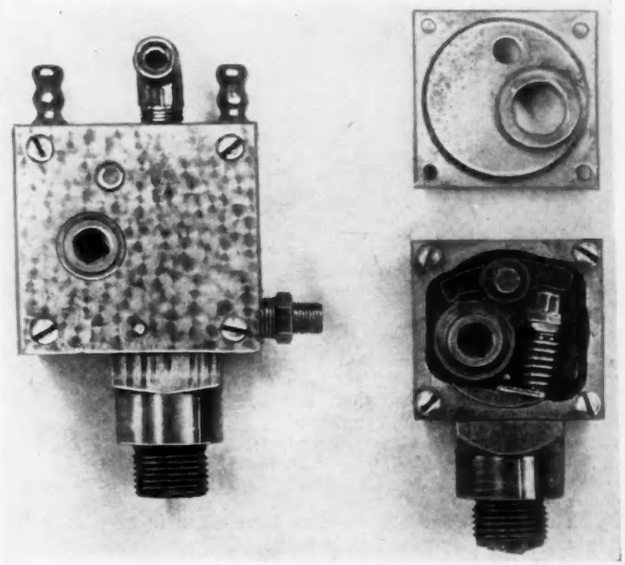


FIG. 9—SAMPLING-VALVE INDICATOR DEVELOPED BY THE MATERIEL DIVISION

This Unit Screws Directly into the Wall of the Combustion Chamber and Makes a Permanent Record on Ordinary Indicator Paper. It Is Particularly Suitable for Use in Connection with Work on Engine Development, Valve Timing, Supercharging and Similar Problems. A Passage in the Valve Body Provides for Cooling-Water Circulation, and a Special Form of Balanced Valve Relieves the Valve Spring of Load Due to Pressure Trapped in the Recording System

The phase gear, shown in Fig. 10, is fitted with a grooved drum adapted to pull the drum cord of a conventional low-speed indicator. Thus, pressure-time diagrams are obtained, one revolution of the indicator drum representing one crankshaft rotation. A simple mechanism can also be fitted to move the drum so as to secure pressure-volume diagrams. However, the pressure-time diagram spreads out the events at top center, both firing and exhaust strokes, so as to enable study of valve timing. After all, little importance attaches to the pressure-volume diagram, except for checking the indicated mean effective pressure, while the study of valve events

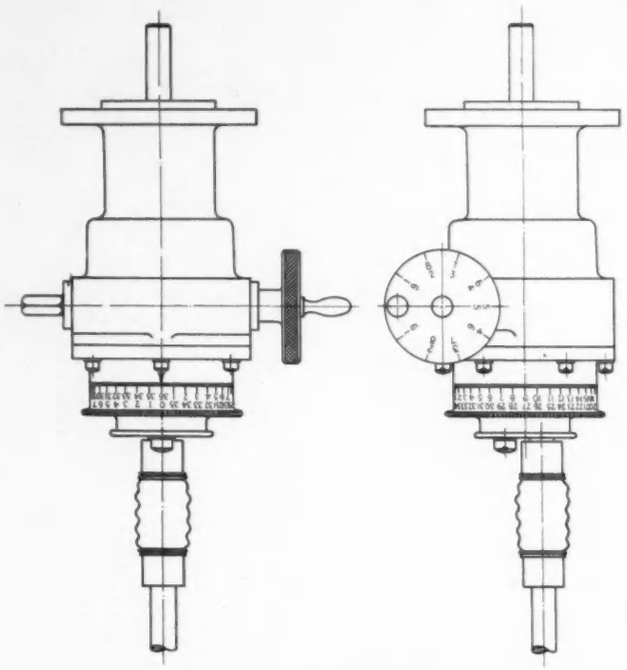


FIG. 10—PHASE GEAR OF THE MATERIEL DIVISION'S SAMPLING-VALVE INDICATOR

This Gear Is Fitted with a Grooved Drum That Is Adapted To Pull the Drum Cord of a Conventional Low-Speed Indicator, thus Enabling Pressure-Time Diagrams To Be Obtained, in Which One Revolution of the Indicator Drum Corresponds to One Rotation of the Crankshaft

by weak-spring pressure-time diagrams is of direct importance in engine research. The pressure-volume mechanism can readily be added to enable taking records of this type if desired.

The sampling-valve method has the advantage, over the balanced-pressure type, of simplicity, not requiring pressure tanks nor vacuum pumps, and it makes a permanent record of satisfactory size on ordinary indicator paper. The recently developed wax-sensitized paper is, in some respects, better for this purpose. It requires very light stylus pressure and has less pencil friction than the metallic sensitized paper. It does not, however, stand handling well, and oil darkens the wax coating. The cost of this paper is very moderate, and it has proved very satisfactory for most purposes.

The Materiel Division sampling-valve was directly compared with the Bureau of Standards balanced-diaphragm indicator by taking readings of pressure and phase angle on a Liberty single-cylinder test engine, using the same gages and manometer as are used on the latter instrument. The pressure diagrams were almost identical when plotted, and it was concluded that the sampling type is inherently as precise as any other known method. The greatest care must be exercised to insure no leaking of pressure on the recording side of the system. The chief disadvantage is that a good installation is not simple to secure, because of the mechanical drive required. For single-cylinder research work, the drive for the phase gear can be built in and a tapped hole conveniently located for the valve.

Discussion of Pressure Records

Fig. 5 shows typical records of cylinder pressure in aircraft engines, when using the carbon-pile indicator. In all but the card at the upper right, the effects of vibration of the pressure element are evident, these oscillations being so bad as to render the cards of little value. The remaining card was taken on a test engine having a cast-iron cylinder, and the very poor record is due to use of an alternating-current arc, since direct current was not available. The form of a detonation pressure-peak can be observed, as well as the great variation in shape and time of pressure peak. This card was made with the pressure-volume mechanism offset 90 deg. In both cards of the upper pair, time is from right to left. The lower cards are typical of the records obtained on aircraft-engine-cylinders, using the pressure-volume mechanism. The effect of vibration obscures the form of the record, sometimes making its interpretation extremely difficult.

The upper pair of cards reproduced in Fig. 6 were taken at the oil-pressure-gage connection on a 12-cylinder aircraft engine. The card at the left was taken at a speed of about 600 r.p.m. and shows the effect of oil-groove register. The other card shows this effect to a small degree and also what appears to be pulsations due to the teeth of the gear-type oil-pump used. The card reproduced at the lower left shows the pressure cycle of a fuel-injection system at low speed, while that at the right shows the pressure cycle at high speed. The waves present in both records appear to be pressure waves in the fuel, since they change radically with tube length and diameter. On the lower cards time is from right to left, due to the direction of rotation of the mirror unit.

Fig. 11 shows typical records obtained with the Materiel Division sampling-valve indicator. The three upper cards at the left were taken on a 12-cylinder V-engine at 2400 r.p.m. In the top card, the engine was operating unsupercharged, while in the one directly below a small boost was used, which is evident in the weak-spring card. The sharp peaks at maximum pressure resulted from incipient detonation, this being the only way in which detonation is shown by this indicator. In the next to the bottom card, an exhaust manifold was used at two back-pressures, and the effects of cylinder interference and back pressure are plainly visible in the record obtained.

The bottom pair of cards show records on a single-cylinder test engine, with poor valve-timing at the left and improved timing at the right. In the latter case, an error evidently was made in the scale of the high-pressure card, which should have been 300 lb. per sq. in. instead of 180 as shown. The peak pressure would then be 425 lb. per sq. in. instead of 245, as indicated. The relatively fat card is due to the 5:1 compression ratio used; and the engine was supercharged, as shown by the low-pressure card.

The three upper cards at the right were taken in the intake pipe close to the intake valve and show typical pressure surges due to resonance and gas inertia. The lowest of these cards was taken at a boost of 13.1 in. of mercury.

Fig. 12 shows some interesting cards taken on a supercharged single-cylinder test engine. At the top left the compression curve is shown. The compression ratio is 4:1, and the indicator was phased by setting to the peak pressure, as indicated. The phase index was marked and thereafter a T.D.C. mark was made on the cards at this point as shown on some of the other cards. The middle left card was taken at a boost of 20 in. of mercury, while that at the bottom shows unsupercharged operation, a 10-in. boost and a 10-in. boost with 10-in. back pressure. Here, too early exhaust-valve closing is shown by the pressure peak at the end of the exhaust stroke. In the top right card, the boost was 20 in., but the peak pressure was low, due to late ignition. The re-

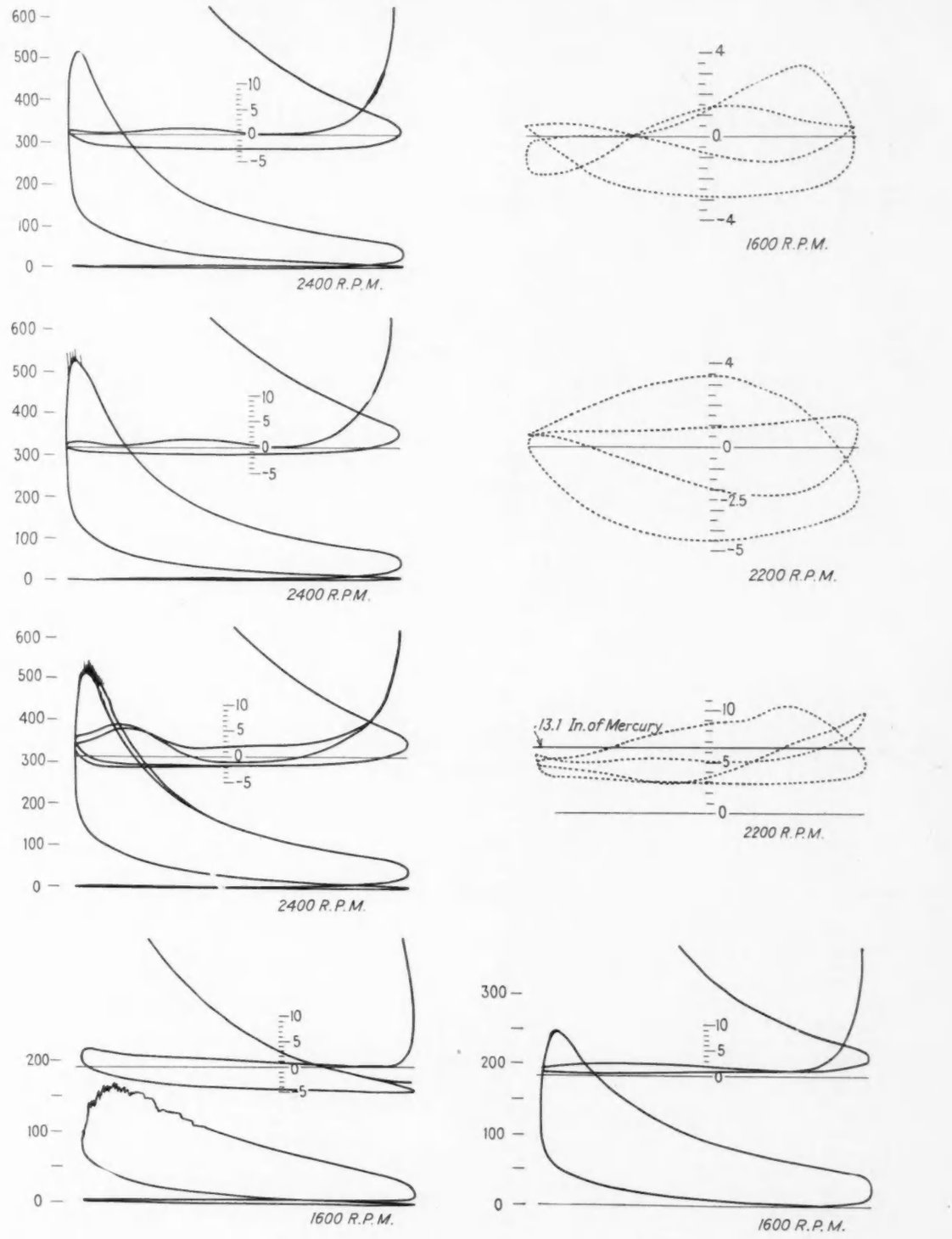
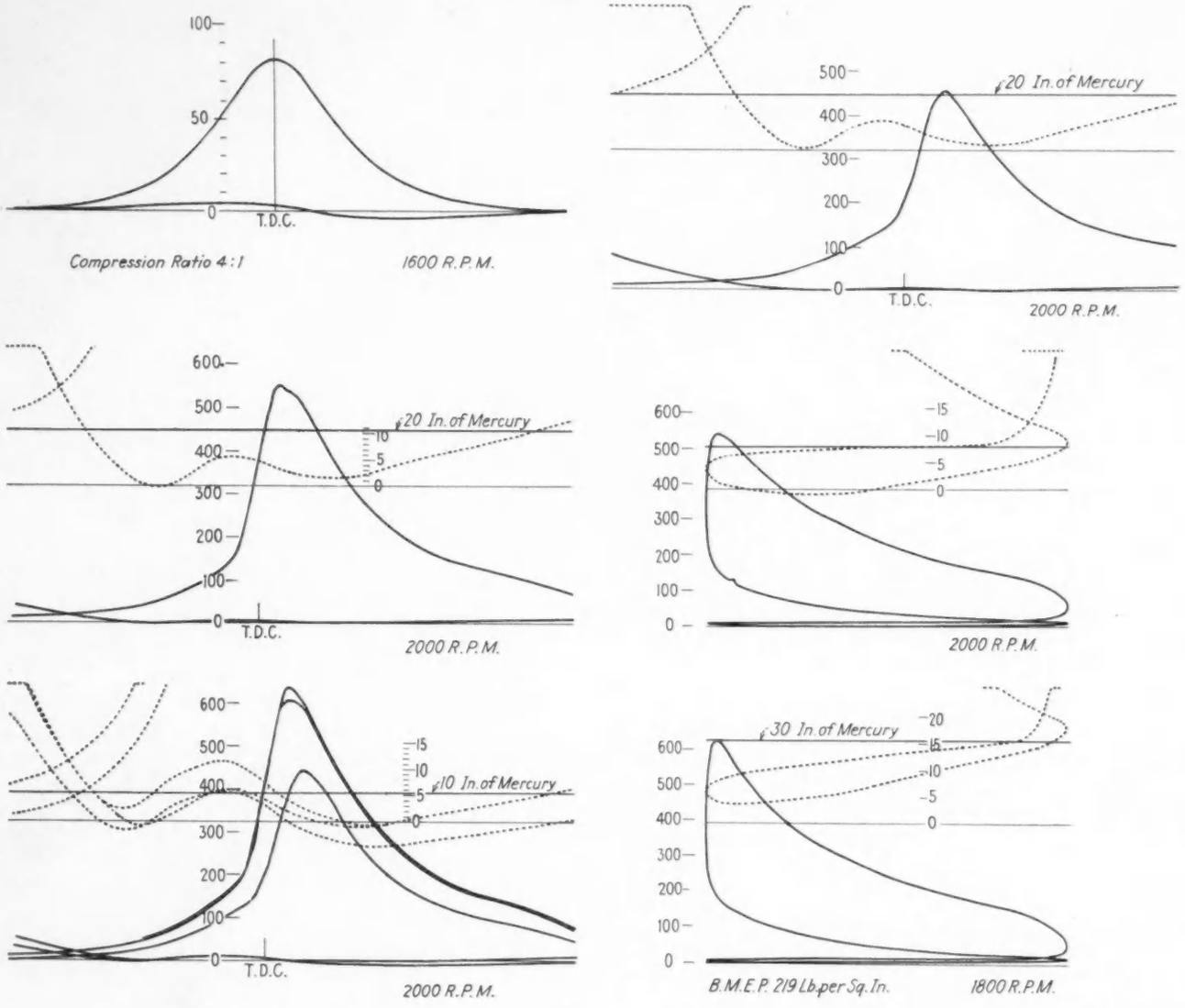


FIG. 11—TYPICAL RECORDS OBTAINED WITH THE MATERIEL DIVISION'S SAMPLING-VALVE INDICATOR

The Upper Three Cards of the Left Row Were Taken on a 12-Cylinder V-Engine at 1200 R.P.M. and Show (Top) Unsurpercharged Operation, (Upper Middle) Operation with a Small Boost and Incipient Detonation as Indicated by the Sharp Peaks at Maximum Pressure, and (Lower Middle) Use of an Exhaust Manifold at Two Back-Pressures. The Upper Three Cards of the Right Row Were Taken in the Intake Pipe Close to the Intake Valve and Show Typical Pressure Surges Due to Resonance and Gas Inertia. The Pair of Cards at the Bottom Show Records Obtained from a Single-Cylinder Test Engine with Poor Valve-Timing (Left) and (Right) After Improvement in the Timing Had Been Obtained



Compression Curve with a 4:1 Compression Ratio
Operation at a Boost of 20 In. of Mercury
Unsupercharged Operation and Operation with a 10-In. Boost and a 10-In. Boost with a Back Pressure of 10 In.

Low Peak Pressure Is Due to Late Ignition with a Boost of 20 In
Operation at a 16-In. Boost
Operation at a 30-In. Boost

FIG. 12—OTHER CARDS TAKEN ON A SUPERCHARGED SINGLE-CYLINDER TEST ENGINE WITH THE INDICATOR SHOWN IN FIG. 9

maining cards at the right are pressure-volume diagrams at 16 and 30-in. boost pressure. The brake mean effective pressure in the card at the bottom right is 219 lb. per sq. in. The source of supercharge was an air compressor, and the boost pressure acting on the intake stroke should have been sufficient to operate the engine; hence the mechanical efficiency of 100 per cent which this card shows by its indicated mean effective pressure of 216 lb. per sq. in. On another occasion, a test engine with a 39-in. boost refused to stop when the ignition and fuel were cut off. Its mechanical efficiency was evidently above 100 per cent, since the dynamometer caused a considerable drag due to the residual field and windage.

Conclusion

From the tests so far conducted, the sampling or averaging type of indicator and the electric or instantaneous type each appears to have fields of usefulness not covered by the other. For transient and qualitative work such as combustion study, the electric indicator will still be the most satisfactory device, while for engine development, valve-timing studies, supercharging and the like the sampling device probably is the most useful. One type is strictly a laboratory instrument, while the other is a most convenient tool for the use of the engine-research man, comparing in usefulness with the steam-engine indicator, as related to steam-engine development.

Journal-Bearing Friction in the Region of Thin-Film Lubrication[1]

Semi-Annual Meeting Paper

By S. A. McKee[2] and T. R. McKee[3]

AUTOMOTIVE engineers, who have to deal with high-speed high-load journal bearings and are confronted with fear of the consequences of "ragged-edge" thin-film lubrication, will find much of interest and value in this report. It covers the information obtained and conclusions reached as a result of the extension of research into the region of thin-film lubrication, in which little investigation has heretofore been made.

Tests made in a four-bearing machine, using both high-tin babbitt and high-lead bronze bearings and operated on lubricants of various viscosities over a wide range of speed and load, showed conclusively that the coefficient of friction depends directly upon the viscosity even in the thin-film region. They indicated defnitely that the friction coefficient is more likely a function of $ZN/\sqrt{P}$ than of ZN/P. It was found also to depend upon the metal of which the bearing is constructed, whereas, in the region of thick-film lubrication, the coefficient of friction is independent of the composition of the bearing metal.

IN recent years there has been considerable experimental evidence to indicate that the concept of a load-carrying oil film forms a sound basis for the design of journal bearings, although the general problem has not been solved completely. Various investigations have determined, not only that a complete film of lubricant between journal and bearing can be built up under proper conditions, but also that the behavior of the bearing follows the same general lines as indicated by theoretical analyses of the mechanics of the film. These investigations, however, have been concerned mainly with the performance of the bearing when the conditions of operation are such that a complete film of the lubricant is maintained.

During the last decade the trend of development in automotive practice has been toward the use of higher speeds and higher pressures, with a consequent necessity for a greater rate of heat dissipation by the bearings and a greater possibility for oil-film breakdown. Because of this trend there has arisen a marked difference of opinion as to whether or not the criterion for bearing design should be based strictly on concepts derived from characteristics typical of fluid-film operation. Realizing the importance of this question, the work at the Bureau of Standards on the study of bearing friction has been extended to include this investigation of the effects of systematic changes in the various factors of operation upon the friction of the bearing when conditions are such that a complete film of the lubricant is not maintained between the rubbing surfaces.

In the study of bearing friction it has been found convenient to indicate graphically the performance of the bearing by plotting the coefficient of friction f against the generalized operating variable ZN/P, where Z is the absolute viscosity of the lubricant, expressed in centipoises; N the speed of the journal, in revolutions per minute; and P the bearing pressure, that is, load per unit projected area, in pounds per square inch. Various publications[4] have indicated the justification for and the advantages of the use of this generalized variable in place of the separate factors of operation.

Fig. 1 shows a typical $[f\text{-}(ZN/P)]$ curve. That part of the curve to the right of the minimum point has been termed the region of "stable"[5] or thick-film lubrication, where a complete film of the lubricant supposedly separates the rubbing surfaces. That part of the curve to the left has been termed the region of "unstable"[5] or

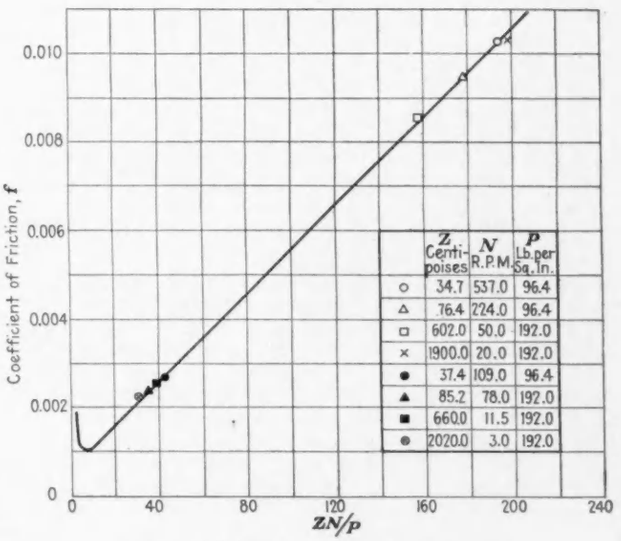


FIG. 1—TYPICAL $[f - (ZN/P)]$ CURVE

[1] Publication approved by the Director of the Bureau of Standards, Department of Commerce.

[2] S.M.S.A.E.—Associate mechanical engineer, Bureau of Standards, City of Washington.

[3] Research associate, Bureau of Standards, City of Washington.

[4] See *Transactions of the American Society of Mechanical Engineers*, vol. 37, p. 167; see also THE JOURNAL, July, 1922, p. 49.

[5] See THE JOURNAL, July, 1922, p. 49.

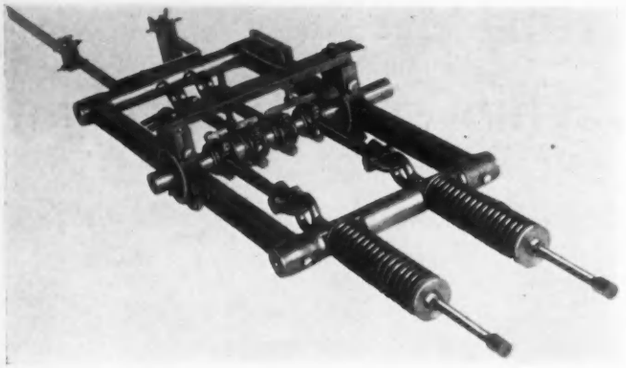


FIG. 2—ESSENTIAL PARTS OF THE JOURNAL-BEARING-FRICTION MACHINE

thin-film lubrication, where the oil film has become so thin that the viscosity, as ordinarily measured, no longer is the sole factor.

The curve in Fig. 1 was obtained from actual experimental data on one set of bearings. It provides an interesting experimental check upon the validity of the assumption that f is a function of ZN/P in the region of thick-film lubrication. It will be noted that the observed data fall within reasonable experimental error upon the same curve, although there is a wide range of variation in the values of the load, speed and viscosity, for the individual points. The same graphical method was chosen in this study to indicate the effects on the friction of changes in the individual operating variables in the region of thin-film lubrication.

[6] See THE JOURNAL, January, 1927, p. 3. Much of the previous experimental work which forms a background for this particular investigation was done in cooperation with the Special Research Committee on Lubrication of the American Society of Mechanical Engineers; see also *Mechanical Engineering*, December, 1927, p. 1335; see also *Transactions of the American Society of Mechanical Engineers*, APM-51-15, p. 161.

TABLE 1—CONSTANT-LOAD RUNS WITH BABBITT BEARINGS

Oil A				Oil B				Oil C			
Speed, R.P.M.	Viscosity Z	f	ZN/P	Speed, R.P.M	Viscosity Z	f	ZN/P	Speed, R.P.M	Viscosity Z	f	ZN/P
					Load, 300.3 Lb.						
22.0	52.0	0.00082	6.00	7.00	147.0	0.00076	5.35	3.30	273.0	0.00074	4.70
18.0	52.0	0.00075	4.88	6.00	146.0	0.00067	4.56	2.80	273.0	0.00069	3.99
10.0	52.0	0.00069	2.70	5.10	147.0	0.00063	3.90	2.10	273.0	0.00057	2.98
7.00	52.0	0.00048	1.90	3.90	147.0	0.00061	2.98	1.65	273.0	0.00050	2.35
5.30	52.0	0.00043	1.43	2.80	147.0	0.00054	2.15	1.40	273.0	0.00048	2.00
3.80	52.0	0.00048	1.03	1.90	147.0	0.00052	1.45	1.00	273.0	0.00068	1.43
3.30	52.0	0.00071	0.89	1.90	143.0	0.00053	1.41	0.85	273.0	0.00096	1.21
2.90	52.0	0.00108	0.78	1.40	143.0	0.00062	1.04	0.55	273.0	0.00306	0.79
.....	.....	.....	.....	1.17	143.0	0.00119	0.88	.....	.....	.....	.....
.....	.....	.....	.....	0.80	143.0	0.00780	0.60	.....	.....	.....	.....
					Load, 200.5 Lb.						
15.8	51.0	0.00082	6.25	3.30	151.0	0.00071	3.88	2.50	273.0	0.00079	5.30
11.3	51.0	0.00070	4.48	2.60	151.0	0.00059	3.06	2.10	273.0	0.00062	4.50
8.60	51.0	0.00060	3.42	2.15	151.0	0.00058	2.53	1.65	273.0	0.00059	3.50
6.25	51.0	0.00055	2.50	1.73	151.0	0.00058	2.03	0.95	273.0	0.00059	2.02
4.85	51.0	0.00049	1.92	1.20	151.0	0.00062	1.41	0.83	273.0	0.00059	1.75
4.05	51.0	0.00048	1.61	0.95	151.0	0.00095	1.13	0.75	273.0	0.00104	1.60
3.35	51.0	0.00052	1.33	0.80	151.0	0.00247	0.94	0.65	273.0	0.00138	1.38
2.60	51.0	0.00068	1.03	0.60	151.0	0.01250	0.71	0.35	273.0	0.00700	0.75
2.35	51.0	0.00112	0.94	.....	.....	.....	.....	.....	.....	.....	.....
2.00	50.9	0.00470	0.79	.....	.....	.....	.....	.....	.....	.....	.....
1.60	50.9	0.00930	0.63	.....	.....	.....	.....	.....	.....	.....	.....
					Load, 100.9 Lb						
8.10	51.1	0.00088	6.40	1.73	149.0	0.00065	3.99	1.70	263.0	0.00085	6.90
6.00	51.1	0.00068	4.70	1.40	149.0	0.00061	3.24	1.20	263.0	0.00078	4.90
4.75	51.1	0.00068	3.76	1.25	149.0	0.00071	2.88	0.78	263.0	0.00068	3.15
3.20	51.1	0.00065	2.52	1.00	148.0	0.00071	2.30	0.60	263.0	0.00065	2.45
2.50	51.0	0.00062	1.97	0.85	147.0	0.00071	1.94	0.50	263.0	0.00075	2.04
1.80	51.0	0.00091	1.42	0.75	147.0	0.00088	1.71	0.40	263.0	0.00156	1.62
1.55	51.0	0.00149	1.23	0.65	147.0	0.00137	1.42	0.30	263.0	0.01060	1.22
1.10	51.0	0.01408	0.87	0.55	147.0	0.00440	1.25	0.24	263.0	0.02300	0.97
					Load, 51.8 Lb.						
2.10	48.9	0.00069	3.10	1.00	143.0	0.00100	4.32	0.70	273.0	0.00088	5.80
1.80	48.9	0.00076	2.66	0.85	143.9	0.00076	3.68	0.60	273.0	0.00088	4.95
1.45	48.9	0.00076	2.14	0.75	143.0	0.00069	3.24	0.50	273.0	0.00101	4.10
1.10	48.9	0.00152	1.62	0.65	143.0	0.00076	2.81	0.40	273.0	0.00088	3.28
0.85	48.9	0.00516	1.26	0.55	143.0	0.00088	2.38	0.35	273.0	0.00114	2.88
0.75	48.9	0.01170	1.11	0.49	143.0	0.00216	2.12	0.30	273.0	0.00203	2.46
.....	.....	.....	.....	0.40	143.0	0.01090	1.73	0.24	273.0	0.00470	1.98
.....	.....	.....	.....	0.25	143.0	0.03300	1.08	0.15	273.0	0.02790	1.24

The Friction Machine and Its Operation

The friction machine available for this work had been used in other investigations and a complete description was given in previous publications[6]. The essential parts of this machine are shown in Fig. 2. It consists of four special bearings, of the same general shape as those of engine connecting-rods, mounted on a steel shaft set in a lathe and held in place by a floating frame which provides a means for measuring the frictional torque. Load is applied by the coil springs fitted to the two end bearings. Oil is fed to the bearings under a pressure of about 1.5 lb. per sq. in. through rubber tubing from a glass bottle. Copper-constantan thermocouples mounted in the second and fourth bearings, in such a manner that the junctions are close to the bearing surfaces, are used for measuring the working temperature. A water-circulating system in conjunction with the hollow test-shaft provides a means for temperature control.

TABLE 2—CONSTANT-SPEED RUNS WITH BABBITT BEARINGS

Load, Lb.	Oil A Viscosity Z	Oil A f	Oil A ZN/P	Oil B Viscosity Z	Oil B f	Oil B ZN/P	Oil C Viscosity Z	Oil C f	Oil C ZN/P
	2.90 R. p. m.			*1.30 R. p. m.*			*0.70 R. p. m.*		
51.8	45.2	0.00061	3.98	151.0	0.00080	5.95	345.0	0.00076	7.30
100.9	45.5	0.00052	2.04	151.0	0.00075	3.04	355.0	0.00072	3.85
150.6	45.5	0.00043	1.37	150.7	0.00065	2.02	360.0	0.00061	2.60
200.5	45.5	0.00053	1.03	149.0	0.00049	1.51	368.0	0.00052	2.00
250.4	45.5	0.00089	0.82	150.7	0.00058	1.22	368.0	0.00070	1.60
300.3	46.2	0.00113	0.70	150.7	0.00057	1.01	368.0	0.00057	1.34
	2.40 R. p. m.			*0.80 R. p. m.*			*0.50 R. p. m.*		
51.8	45.2	0.00057	3.28	193.0	0.00076	4.66	352.0	0.00089	5.30
100.9	44.2	0.00078	1.65	187.0	0.00065	2.32	357.0	0.00104	2.76
150.6	44.2	0.00104	1.10	188.0	0.00126	1.56	350.0	0.00097	1.82
200.5	44.2	0.00144	0.83	186.0	0.00160	1.16	350.0	0.00118	1.36
250.4	44.2	0.00305	0.66	186.0	0.00233	0.93	350.0	0.00178	1.09
300.3	44.2	0.00324	0.55	186.0	0.00300	0.78	350.0	0.00183	0.91
	2.00 R. p. m.			*0.55 R. p. m.*			*0.40 R. p. m.*		
51.8	41.3	0.00076	2.50	186.0	0.00082	3.09	350.0	0.00076	4.20
100.9	42.4	0.00084	1.32	186.0	0.00195	1.59	350.0	0.00208	2.18
150.6	41.5	0.00412	0.86	186.0	0.01020	1.06	350.0	0.00240	1.45
200.5	41.5	0.00630	0.65	186.0	0.01300	0.80	350.0	0.00280	1.09
250.4	41.5	0.00800	0.51	186.0	0.01480	0.63	350.0	0.00307	0.87
300.3	41.5	0.01200	0.43	186.0	0.01580	0.54	350.0	0.00321	0.73

The Journal and Bearings Tested

The shaft used in these tests was made of a high-carbon-tungsten tool-steel, heat-treated, ground and lapped. Its heat-treatment was: heated at 1500 deg. fahr., held for 30 min., quenched in oil, reheated at 850 deg. fahr. for 45 min., and air-cooled. It had a Brinell hardness of about 179. Its average diameter at the journals was 1.2504 in. It had been used in previous work and the journals were in a well-run-in condition.

Two types of bearing metals were tested: a high-tin babbitt and a high-lead bronze. The babbitt bearings were made up of solid bronze sleeves with a lining of babbitt about 1/16 in. thick. The bronze sleeves were tinned in a bath of pure tin at a temperature of 620 deg. fahr. The babbitt metal was centrifugally cast at a temperature of 940 deg. fahr. and a speed of 1150 r.p.m. The average of analyses of four samples of the babbitt taken after casting was 4.38 per cent copper, 90.40 per cent tin, 0.49 per cent lead and 4.73 per cent antimony. The bearings were faced off square to pro-

vide a bearing length of 1.250 in. and machined to size by the manufacturer. The average diameter of the set of four bearings was 1.2515 in., making an average running clearance, that is, difference of diameters, of 0.0011 in. and a clearance-diameter ratio of about 0.0009.

The bronze bearings were made up into solid sleeves. The approximate composition of the metal used was 72 per cent copper, 3 per cent tin and 25 per cent lead. They were faced off to a length of 1.250 in. and broached to size by the manufacturer. Their average diameter was 1.2519 in., making an average clearance of 0.0015 in. and a clearance-diameter ratio of about 0.0012.

Each bearing had a single oil-hole at the center of the unloaded side and had no oil grooves or relieved spaces.

Description of the Lubricants Used

Both sets of bearings were tested with three mineral oils having viscosities of 75, 139 and 230 sec. Saybolt Universal at 130 deg. fahr. The average running temperature was about 75 deg. fahr., the absolute viscosities of these oils at this temperature being 44, 129 and 246 centipoises, respectively. For convenience, the 75-sec. oil is designated as Oil *A*, the 139-sec. oil as Oil *B*, and the 230-sec. oil as Oil *C*.

[1] See *Mechanical Engineering*, December, 1927, p. 1335.

TABLE 3—CONSTANT-LOAD RUNS WITH BRONZE BEARINGS

Oil A				Oil B				Oil C			
Speed, R.P.M	Viscosity Z	f	ZN/P	Speed, R.P.M	Viscosity Z	f	ZN/P	Speed, R.P.M	Viscosity Z	f	ZN/P
				Load, 300.3 Lb.							
250.0	42.9	0.00286	55.6	97.0	117.0	0.00290	59.0	50.0	211.0	0.00286	55.0
209.0	42.9	0.00254	46.6	85.0	117.0	0.00270	52.0	42.0	211.0	0.00260	46.0
156.0	43.1	0.00214	35.0	74.0	117.0	0.00252	45.0	34.0	214.0	0.00225	38.0
109.0	43.1	0.00178	24.4	58.5	118.0	0.00228	36.0	28.0	214.0	0.00209	31.2
76.0	42.9	0.00150	17.0	46.5	118.0	0.00194	28.6	16.5	214.0	0.00160	18.4
59.0	42.9	0.00128	13.1	30.0	119.1	0.00157	18.7	11.5	214.0	0.00139	12.7
41.0	42.9	0.00108	9.14	18.0	119.1	0.00130	11.2	8.00	214.0	0.00113	8.90
25.0	42.9	0.00097	5.60	12.0	119.1	0.00104	7.50	5.00	214.0	0.00104	5.50
21.8	42.9	0.00091	4.82	8.00	119.1	0.00099	4.95	3.60	214.0	0.00100	4.00
19.0	42.9	0.00087	4.30	6.00	116.0	0.00101	3.62	3.15	214.0	0.00102	3.50
17.8	42.9	0.00087	3.96	5.40	116.0	0.00123	3.28	2.85	215.0	0.00109	3.18
16.0	42.9	0.00100	3.58	4.00	116.0	0.00250	2.42	2.60	215.0	0.00146	2.91
14.3	42.9	0.00143	3.18	3.10	116.0	0.00442	1.86	1.85	215.0	0.00422	2.06
12.8	42.9	0.00274	2.86	2.50	116.0	0.00940	1.51	1.50	215.0	0.00732	1.68
10.0	42.9	0.00820	2.23	.....	.....	.....	.....	.....	.....	.....	.....
8.00	42.9	0.01220	1.78	.....	.....	.....	.....	.....	.....	.....	.....
				Load, 200.5 Lb							
70.0	41.5	0.00173	22.6	63.0	113.0	0.00294	55.5	25.0	202.0	0.00228	39.4
53.5	41.5	0.00153	17.3	50.0	113.0	0.00258	44.5	16.5	202.0	0.00190	26.0
43.5	42.3	0.00140	14.4	43.0	113.0	0.00240	38.0	10.5	202.0	0.00151	16.5
32.0	43.1	0.00118	10.8	30.0	113.0	0.00189	26.5	7.75	202.0	0.00123	12.3
25.5	43.1	0.00118	8.60	20.0	113.0	0.00151	17.6	4.50	202.0	0.00105	7.10
23.0	43.1	0.00118	7.70	15.0	113.5	0.00130	13.3	3.30	202.0	0.00105	5.20
20.0	43.1	0.00118	6.70	10.0	113.5	0.00124	8.85	3.10	202.0	0.00114	4.90
18.0	43.1	0.00120	6.05	7.00	113.5	0.00095	6.20	2.80	202.0	0.00170	4.40
15.0	43.1	0.00120	5.01	5.30	113.5	0.00097	4.70	2.30	202.0	0.00258	3.60
12.5	43.1	0.00127	4.20	4.60	113.5	0.00176	3.28	2.00	202.0	0.00492	3.15
11.0	43.1	0.00160	3.70	3.30	113.5	0.00242	2.93	1.50	202.0	0.00665	2.36
9.00	43.1	0.00400	3.15	2.30	113.5	0.00953	2.02	1.15	202.0	0.01350	1.80
7.00	43.1	0.01010	2.36	2.00	113.5	0.01386	1.78	.....	.....	.....	.....
				Load, 100.9 Lb.							
44.0	43.1	0.00195	29.3	32.5	118.0	0.00304	59.5	15.3	226.0	0.00292	53.0
30.0	43.1	0.00162	20.0	22.5	118.0	0.00247	41.2	11.9	226.0	0.00239	41.5
24.0	43.1	0.00136	16.0	17.0	118.0	0.00207	31.0	9.00	226.0	0.00224	31.6
20.0	43.1	0.00130	13.3	12.5	118.0	0.00162	22.8	7.25	226.0	0.00200	25.4
18.0	43.1	0.00130	12.0	6.00	118.0	0.00122	10.9	5.00	226.0	0.00169	17.5
16.0	43.1	0.00130	10.7	4.75	118.0	0.00110	8.75	4.00	226.0	0.00150	14.0
12.0	43.1	0.00118	8.00	3.60	115.0	0.00104	6.40	3.00	226.0	0.00142	10.4
9.00	43.1	0.00156	6.00	3.20	115.0	0.00104	5.70	2.00	226.0	0.00117	7.00
7.00	43.1	0.00300	4.65	2.70	113.0	0.00110	4.72	1.65	226.0	0.00137	5.80
6.10	43.1	0.00358	4.10	2.30	113.0	0.00207	4.00	1.45	226.0	0.00214	5.10
5.50	43.1	0.00600	3.66	2.00	113.0	0.00418	3.50	1.22	226.0	0.00300	4.30
.....	.....	.....	.....	1.65	113.0	0.01300	2.88	1.10	226.0	0.00340	3.86
.....	.....	.....	.....	1.30	113.0	0.02640	2.26	0.80	226.0	0.01100	2.80
.....	.....	.....	.....	1.15	113.0	0.03520	2.00	0.60	226.0	0.02900	2.10
				Load, 51.8 Lb.							
42.5	41.5	0.00279	53.5	20.0	110.0	0.00318	67.5	11.0	223.0	0.00318	74.0
32.5	42.3	0.00240	41.6	16.0	110.0	0.00264	53.4	9.00	223.0	0.00290	61.0
24.5	42.3	0.00214	31.0	12.3	110.0	0.00240	40.8	7.00	223.0	0.00254	47.0
20.0	42.3	0.00201	25.6	9.50	109.0	0.00177	31.2	5.50	223.0	0.00228	37.0
16.5	42.8	0.00201	21.2	8.00	109.0	0.00177	26.2	4.50	223.0	0.00214	30.0
12.3	42.8	0.00177	15.9	3.80	109.0	0.00139	12.5	3.60	218.0	0.00214	23.8
9.00	42.8	0.00177	11.6	2.80	109.0	0.00131	9.30	2.70	218.0	0.00189	17.7
7.00	42.8	0.00240	9.10	2.45	109.0	0.00227	8.10	2.20	218.0	0.00183	14.5
5.50	41.6	0.00366	6.90	2.15	109.0	0.00279	7.10	1.80	218.0	0.00200	12.0
4.50	41.6	0.02040	5.70	1.55	109.0	0.00640	5.10	1.40	218.0	0.00164	9.30
3.50	41.6	0.03680	4.40	1.20	109.0	0.01320	3.96	1.10	218.0	0.00228	7.30
.....	.....	.....	.....	1.00	109.0	0.01680	3.28	0.90	218.0	0.00405	5.95
.....	.....	.....	.....	.....	.....	.....	.....	0.70	218.0	0.00760	4.62
.....	.....	.....	.....	.....	.....	.....	.....	0.55	218.0	0.01510	3.62

TABLE 4—CONSTANT-SPEED RUNS WITH BRONZE BEARINGS

Load, Lb.	Oil A Viscosity Z	Oil A f	Oil A ZN/P	Oil B Viscosity Z	Oil B f	Oil B ZN/P	Oil C Viscosity Z	Oil C	Oil C ZN/P
	11.0 R. p. m.			*4.75 R. p. m.*			*2.40 R. p. m.*		
51.8	42.3	0.00152	14.0	111.0	0.00152	16.0	220.0	0.00164	15.9
100.9	42.3	0.00118	7.22	113.5	0.00104	8.40	220.0	0.00117	8.20
150.6	41.6	0.00096	4.76	113.0	0.00092	5.62	223.0	0.00109	5.55
200.5	41.2	0.00091	3.54	110.8	0.00082	4.12	223.0	0.00095	4.16
250.4	41.6	0.00147	2.84	110.0	0.00130	3.28	226.0	0.00108	3.38
300.3	41.6	0.00178	2.39	110.0	0.00158	2.70	226.0	0.00113	2.84
	9.0 R. p. m.			*2.50 R. p. m.*			*1.60 R. p. m.*		
51.8	43.7	0.00137	11.9	112.0	0.00126	8.50	228.0	0.00127	11.1
100.9	43.7	0.00123	6.15	112.0	0.00104	4.37	228.0	0.00123	5.68
150.6	42.3	0.00100	4.00	112.0	0.00250	2.92	228.0	0.00104	3.80
200.5	41.6	0.00160	2.92	112.0	0.00465	2.18	228.0	0.00176	2.84
250.4	41.6	0.00324	2.34	112.0	0.00870	1.74	228.0	0.00243	2.28
300.3	41.6	0.00580	1.96	112.0	0.01100	1.45	228.0	0.00325	1.90
	7.0 R. p. m.			*2.10 R. p. m.*			*1.20 R. p. m.*		
51.8	41.1	0.00101	8.70	109.0	0.00139	6.90	223.0	0.00127	8.10
100.9	41.1	0.00169	4.50	109.0	0.00265	3.56	225.0	0.00162	4.25
150.6	41.1	0.00343	2.99	109.0	0.00900	2.38	226.0	0.00360	2.82
200.5	41.1	0.00607	2.26	109.0	0.01390	1.78	226.0	0.00430	2.12
250.4	41.2	0.00740	1.78	109.0	0.01450	1.43	226.0	0.00618	1.66
300.3	41.3	0.00900	1.50	109.0	0.01560	1.19	226.0	0.00772	1.41

Procedure in Testing

Before the test runs on each set of bearings were started, the bearings were run in until they had reached a practically constant state; that is, the effect of running in[1], in reducing the value of ZN/P at the point of minimum friction, approached zero.

Each test run was made in the decreasing order of ZN/P, starting at a value greater than that at the point of minimum friction. In all, 42 test runs were made; 7 with each of the six combinations of bearings and lubricants. Of these, four were made with different constant loads, using various speeds; and three with different constant speeds, using various loads. The numerical data obtained in these tests are given in Tables 1, 2, 3 and 4.

Discussion of Results

The results of the tests at constant loads on the babbitt bearings using Oil *A* as the lubricant are shown graphically by the $[f\text{-}(ZN/P)]$ curves in Fig. 3, and the curves for the tests at constant speeds with the same bearings and lubricant in Fig. 4. Similar curves showing the results of the tests with the bronze bearings using Oil *A* are shown in Figs. 5 and 6, respectively.

Effects of Change in Load.—From the constant-load curves shown in Fig. 3 it will be noted that, at the higher values of ZN/P, all the data tend to fall along a single line that is representative of the friction of these bearings at any load within the range covered. At the lower values of ZN/P, however, the data for the respective loads tend to form separate curves, the minimum points of the curves for 50, 100, 200 and 300-lb. loads occurring at values of ZN/P of about 2.5, 2.0, 1.3 and 1.0 respectively. It will be noted also from the tabulated data that a similar effect is shown by the tests where different lubricants or different bearings were used. Thus, within the scope of these tests, there is a general indication that, for a given bearing and lubricant, an increase in load tends to decrease the value of ZN/P at

the point of minimum f. Also, for a given value of ZN/P at the left of the point of minimum friction in the region of thin-film or unstable lubrication, an increase in load tends to decrease the coefficient of friction.

Effects of Change in Speed.—The constant-speed curves shown in Fig. 4, together with the data for the other constant-speed tests, indicate a similar general trend that for a given bearing and lubricant an increase in speed tends to decrease the value of ZN/P at the point of minimum f, and for a given value of ZN/P in the region of thin-film lubrication an increase in speed tends to decrease the coefficient of friction.

Effects of Changes in Viscosity.—The three lubricants used in these tests differed considerably in absolute viscosity at the temperature of test, the ratio of the viscosity of Oil C to that of Oil A being approximately 5:1. Thus, a means for estimating the effects of changes in viscosity is provided by comparing the curves for a given bearing at a given constant load with each of the three oils.

A typical example of such a comparison is given by the curves in Fig. 7. From this comparison it is at once apparent that the effects of changes in viscosity are not as definitely and systematically indicated as were the effects of changes in load and speed. The data for the babbitt bearings might be taken as an indication of a small systematic change due to change in viscosity; on the other hand, some of the data for the bronze bearings at the same load but with different oils are practically coincident, while others place the effects of viscosity in the opposite order to that indicated by those for the babbitt metal. It should be noted also that all the tests with a given oil on a given set of bearings were made before another oil was used; thus a direct comparison of the curves for different oils at a given load also involves the possibility of change in the bearing surface because of the number of tests that were made between the particular tests under consideration. Thus it does not appear justifiable to consider these tests as being a

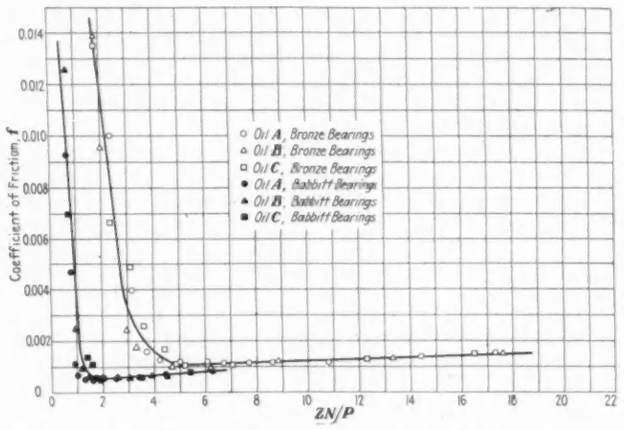
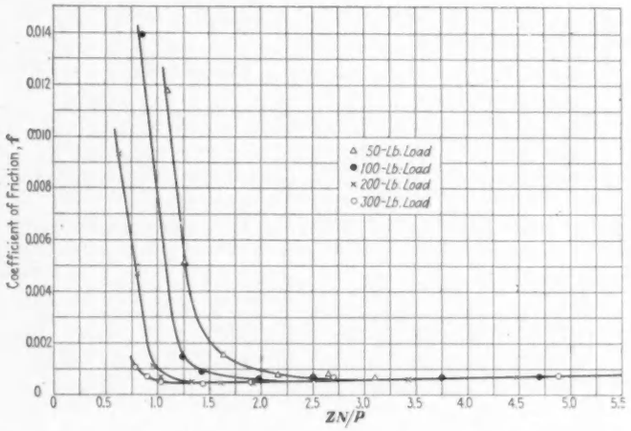
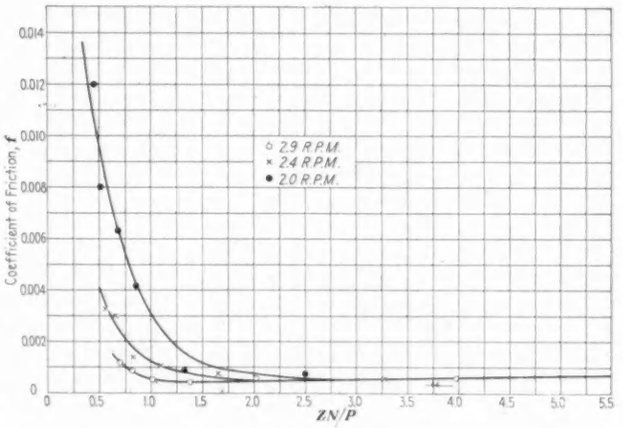
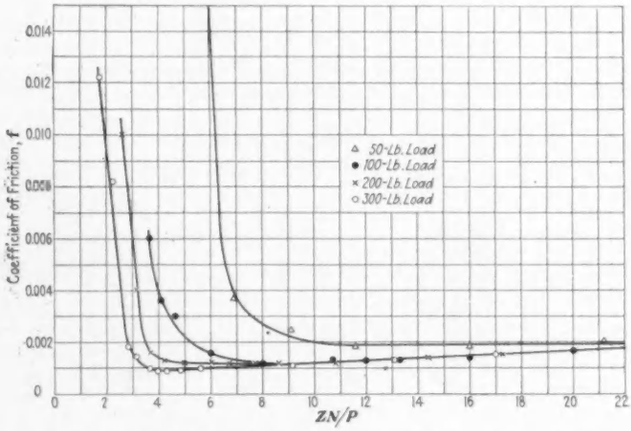


FIG. 7—JOURNAL-FRICTION CURVES AT A 200-LB. LOAD, FOR BABBITT AND BRONZE BEARINGS, USING OILS *A*, *B* AND *C*







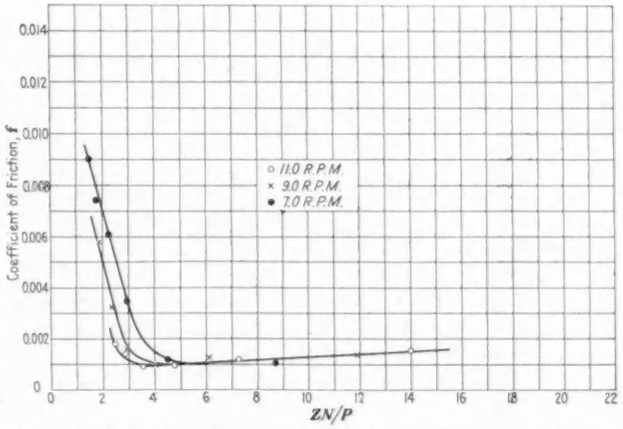


FIG. 3—JOURNAL-FRICTION CURVES AT CONSTANT LOADS FOR BABBITT BEARINGS, USING OIL *A*

FIG. 4—JOURNAL-FRICTION CURVES AT CONSTANT SPEEDS, FOR BABBITT BEARINGS, USING OIL *A*

FIG. 5—JOURNAL-FRICTION CURVES AT CONSTANT LOADS, FOR BRONZE BEARINGS, USING OIL *A*

FIG. 6—JOURNAL-FRICTION CURVES AT CONSTANT SPEEDS, FOR BRONZE BEARINGS, USING OIL *A*

definite indication of a systematic effect of a change in viscosity upon the $[f\text{-}(ZN/P)]$ curve for a given bearing operating with a given load. Within their scope, however, they do tend to indicate that, if such an effect exists, it is relatively small as compared with the effects of changes in load and speed.

Effect of Bearing Metal.—From the descriptions previously given it will be noted that the two sets of bearings were essentially alike except for the composition of the bearing metal; their lengths were the same, the average clearance of each was not markedly different, both had been well run in, and both were run on the same journal. A comparison of the constant-load curves for both sets when using the same lubricant, therefore, ought to give an indication of the difference in the frictional characteristics of the two bearing metals.

Such a comparison is given in Fig. 8, where the $[f\text{-}(ZN/P)]$ curves for the 200-lb. load using Oil *B* are

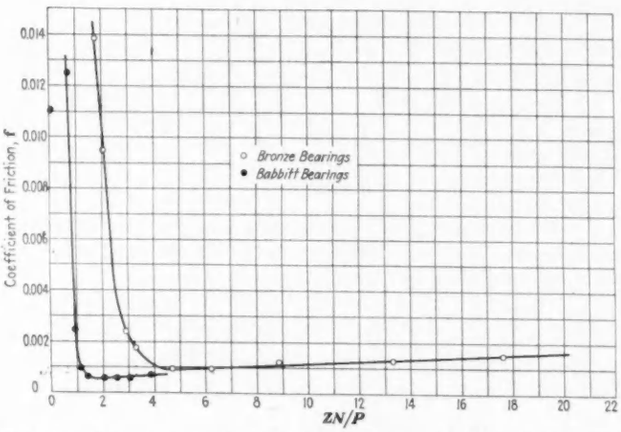


FIG. 8—JOURNAL-FRICTION CURVES AT A 200-LB. LOAD, FOR BABBITT AND BRONZE BEARINGS, USING OIL *B*

plotted for both sets of bearings. From this figure and the data for tests at other loads, it is at once apparent that, from the standpoint of friction alone, the babbitt-metal bearings were superior to the bronze in that at all the loads the values of ZN/P at minimum friction for the babbitt bearings were lower than those for the bronze bearings at corresponding loads. It is of interest to note also that the curves in Figs. 3 and 5 give some indication that the magnitude of a change in the $[f\text{-}(ZN/P)]$ curve caused by a given change in the load is apparently dependent upon the absolute value of ZN/P at which the change occurs. This apparent effect, however, may be due solely to the differences in the characteristics of the two metals.

Running-In Characteristics of Bronze and Babbitt.—During the running-in period with the bronze bearings, sufficient data were obtained for a comparison of the running-in characteristics of these bearings with similar information on babbitt bearings obtained in a previous investigation[8].

The same methods for showing the effect of running in were used in both investigations; namely, to obtain sufficient data for a comparison of the $[f\text{-}(ZN/P)]$ curves for a given set of bearings with different degrees of running in. Curves for the bronze bearings at various stages of running in are shown in Fig. 9. From these curves it is evident that the chief effect of running in is the decrease in the value of ZN/P at the point of minimum friction due to continued running in.

The relative differences in the running-in characteristics of the two types of bearings are given by the curves in Fig. 10, where the total frictional work per bearing[8]

[8] See *Mechanical Engineering*, December, 1927, p. 1335.

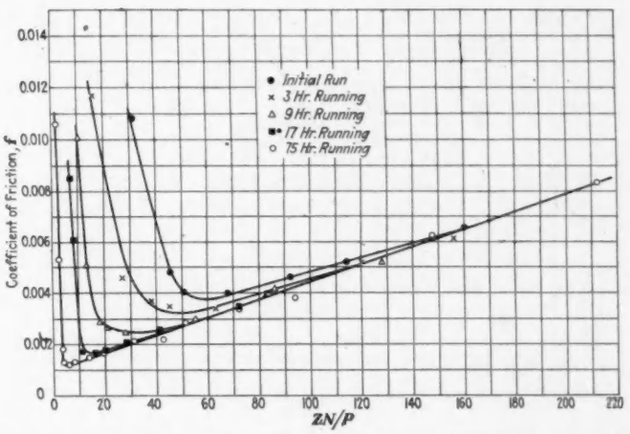


FIG. 9—JOURNAL-FRICTION CURVES, FOR BRONZE BEARINGS, SHOWING THE PROGRESSIVE AMOUNTS OF RUNNING IN

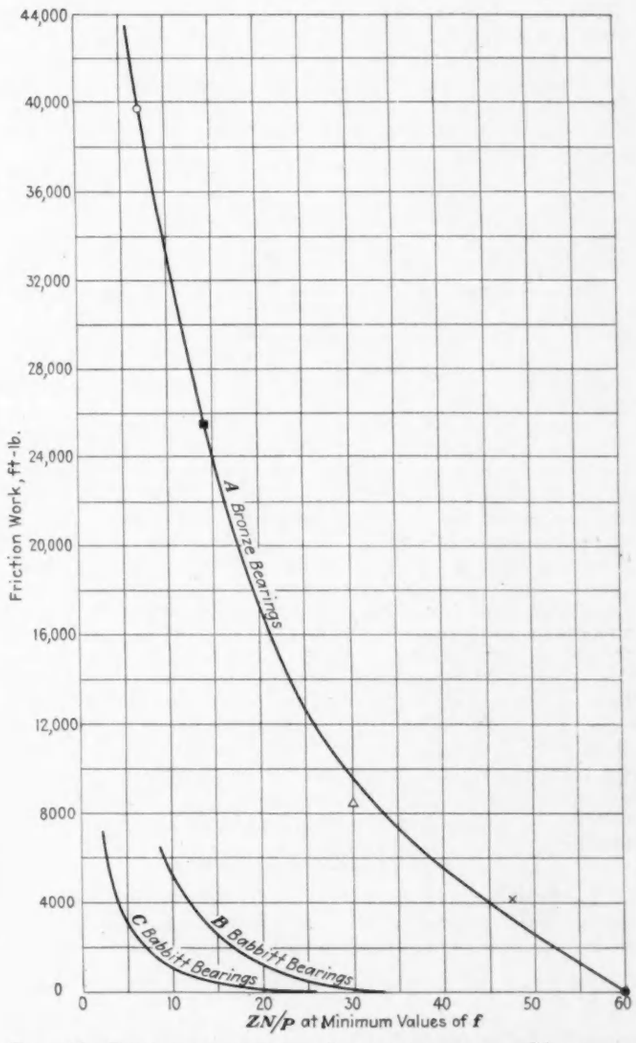


FIG. 10—RELATION BETWEEN LOCATION OF THE MINIMUM POINT AND THE FRICTIONAL WORK DONE ON BABBITT AND ON BRONZE BEARINGS

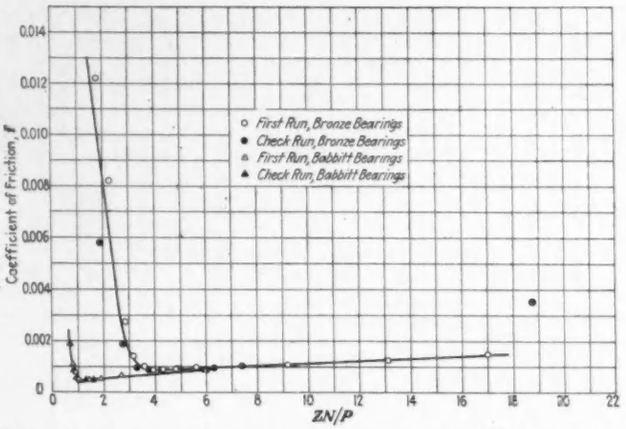


FIG. 11—JOURNAL-FRICTION CURVES AT A 300-LB. LOAD, USING BRONZE AND BABBITT BEARINGS AND SHOWING THE EFFECTS OF CHANGES IN BEARING SURFACE

when operating in the region of thin-film lubrication is plotted against the value of ZN/P at the point of minimum friction. While the curves in Fig. 10 do not take into account the effect of various factors, such as the possibility that a given amount of frictional work when operating at one load and speed may have a different smoothing effect on the bearings than the same amount of work when operating at other loads and speeds, the differences shown are sufficiently marked to indicate that considerably less work was required to run in the babbitt bearings.

Effect of Changes in Bearing Surface.—While both sets of bearings were well run in before the tests were started, throughout a considerable portion of each test run the bearings were operating in the region of thin-film lubrication with the likelihood of there being a certain amount of metallic contact and a consequent tendency to change the condition of the bearing surfaces. It would be expected, therefore, that the condition of the surfaces might vary sufficiently from time to time during the test runs to have a measurable effect upon the friction.

To obtain an estimate of the approximate magnitude of these effects, in some of the constant-load series with a given bearing and lubricant the first run was repeated after the other runs of the series had been completed. Typical examples of the results obtained with both sets of bearings are given in Fig. 11. These curves indicate that any change of friction due to changes in the condition of the bearing surfaces was relatively small as compared with the effects caused by the change in load and speed indicated by the curves in Figs. 3 and 4. It should be mentioned also that, in nearly all of the tests, the order of running was such that the effects shown in Fig. 11 would tend to bring the curves in a given figure closer together rather than to separate them. Thus the differences indicated by the curves in Figs. 3 and 4 are not due to changes in the condition of the surfaces of the bearings, although some of the curves probably are subject to slight corrections to take account of these changes.

Summary and Conclusions

The frictional characteristics of journal bearings in the region of thin-film lubrication indicated by this investigation may be summarized as follows:

(1) The coefficient of friction of a journal bearing in the region of thin-film lubrication is not a function of the generalized operating variable ZN/P alone but is also dependent upon the particular values of the individual operating variables.

(2) For a given bearing operating with a given lubricant, an increase in load or speed tends to decrease the value of ZN/P at the point of minimum friction.

(3) In the region of thin-film lubrication, an increase in load with a proportional increase in speed to maintain a constant value of ZN/P tends to reduce the coefficient of friction of a given bearing operating with a given lubricant.

(4) The composition of the bearing metal has a marked effect upon the frictional characteristics of bearings in the region of thin-film lubrication, the point of minimum friction for a babbitt bearing occurring at a considerably lower value of ZN/P than that for a bronze bearing of similar proportions operating with the same lubricant and at the same load.

While the foregoing characteristics of course apply directly only to the particular conditions present, the tests are, it is believed, of sufficient scope to be an indication of the possibility of a general trend in the frictional characteristics of journal bearings in the region of thin-film lubrication which is possibly of considerable practical significance, especially for high-speed high-pressure-bearing operation typical of present-day automotive practice.

Further steps toward a better understanding of the frictional characteristics of journal bearings in the region of thin-film lubrication suggested by these tests are to measure bearing friction under conditions suitable for a study of the following factors: changes in bearing diameter, length and clearance, and wider ranges of variation in load, speed, viscosity and types of bearing metals and lubricants.

Acknowledgment is made to the Bohn Aluminum & Brass Corp. and to the Bunting Brass & Bronze Co. for furnishing the bearings used in these tests.

APPENDIX

CORRELATION OF EXPERIMENTAL DATA

In the previous investigation[9] on the study of bearing friction in the region of thick-film lubrication, it was found that the coefficient of friction for small-bore journal bearings was represented with reasonable accuracy by

$$f = \Delta f + b\,[(ZN/P)\ (D/C)] \qquad (1)$$

where f is the coefficient of friction; b is a dimensionless constant involving the units employed; Δf was interpreted as a length-diameter ratio-correction term, which was found to be constant for a given length-diameter ratio and to have practically the same value for all

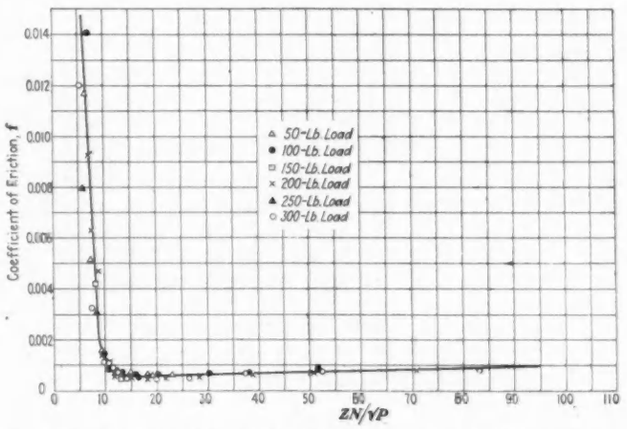


FIG. 12—TYPICAL $[f - (ZN/\sqrt{P})]$ CURVE, FOR BABBITT BEARINGS, USING OIL *A*

[9] See *Transactions of the American Society of Mechanical Engineers*, APM-51-15, p. 161.

ratios above 0.75; D is the journal diameter and C is the running clearance. This equation was considered applicable provided ZN/P is greater than 100 and f is greater than 0.01.

Analysis of the data given in Tables 1, 2, 3 and 4 indicated that, if f is plotted against $ZN/\sqrt{P}$, all the data for a given bearing fall, within reasonable limits, upon the same curve. This curve is shown in Fig. 12, which includes all the data shown in Figs. 3 and 4. All of the data for both bearings are plotted in Fig. 13.

Accordingly, an equation to represent bearing friction in both the stable and the unstable regions of lubrication presumably would be of the form

$$f = \phi\ (ZN/\sqrt{P}) + b\ [(ZN/P)\ (D/C)] \qquad (2)$$

where $\phi\ (ZN/\sqrt{P})$ approaches a constant value at high values of ZN/P. It should be noted, however, that the expression $ZN/\sqrt{P}$ involves the dimensions of the bearing, so that further study would be necessary before an equation of this type could be considered general.

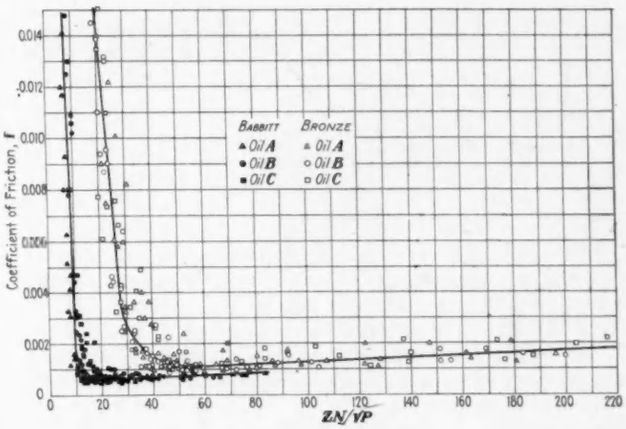


FIG. 13—TYPICAL $[f - (ZN/\sqrt{P})]$ CURVES FOR BOTH BRONZE AND BABBITT BEARINGS, USING OILS *A*, *B* AND *C*

Air-Transport Design and Testing

(*Concluded from p. 360*)

balance, must be provided, and the pilot must work all the time with a device that is really needed only for special purposes.

From this it follows that the engine thrust-line has other than purely aerodynamic implications. Where, for any reason, the thrust line is very low, the nose will drop severely—a condition that is accentuated by great longitudinal stability—when the engines are cut out or fail. Therefore, when necessity for much throttle variation exists, such as approaching to land in bad weather at night, this big load-variation must be taken, and taken easily, on the elevators. It is totally insufficient to demonstrate that this can be effected by the stabilizer adjustment, since the time element and the transport pilot's other preoccupations prevent its effective use. Too high a thrust line is not so troublesome, since any increase in stability opposes the variation caused by the thrust line; but it may decrease engine accessibility for maintenance and must be effectively examined and tested.

The angle of the wing with the fuselage is of importance in landing conditions. Where this is too small, as is often the case, an accurate landing, with the wing at the stalling angle or just above it, will result in the tail wheel touching first and jerking the main wheels onto the ground, unnecessary strain on the structure and discomfort to the passengers. If a "three-point" landing is made, then the wings cannot be at their maximum lift and the landing speed is unnecessarily high. Moreover, at speeds just below landing speed, while the plane is still moving too fast for the application of brakes, the drag on the wings is insufficient for pulling up, and the landing run is greatly increased. Few, if any, objections exist to making the angle of the wings on the ground appreciably greater than the stalling angle. A possible objection is that in cruising flight the fuselage may fly too tail high and have too much drag. This need not cause serious difficulty and can be met in the fuselage and landing-gear design. In any case this matter gives much scope for examination by the test pilot.

The author has suggested a few points showing where test flying should be used to examine the plane beyond merely flying it to see if it is "good enough." Naturally much more could be said on the subject without exhausting it.

The test pilot's recommendations usually can be divided into three classes; namely,

(1) Those changes which ought to be made definitely before the plane is allowed to go into service

(2) Highly desirable, but not quite necessary, changes which are or are not to be made according to the commercial situation

(3) Changes involving serious alteration of general design which must, according to executive decision, hold up the plane until they are made, or must be noted for change in subsequent production.

Finally, the surest method of wasting time and money is to try to save time and money by putting a plane into any sort of production before it is completely and thoroughly tested. And the first plane of any series should be designed so as to permit of easy changes to those parts, such as controls and stabilizing surfaces, which are almost certain to require modification for best results.

Engine Cooling

By Louis Schwitzer[1]

Milwaukee Section Paper

FROM 25 to 35 per cent of the heat energy of the fuel inducted into the cylinders of an internal-combustion engine must be eliminated by the cooling system. As this waste requires the expenditure of energy, the devising of an efficient cooling system is imperative. The author, who is a leading American authority on engine problems, discusses the theory of liquid-cooling, gives heat-transfer and temperature-balance equations that must be satisfied and points out the three interrelated variable factors that must be incorporated in their most economical relation. The cooling system must be studied as a whole, rather than from the standpoint of any particular unit. Some commonly held beliefs regarding fans, fuel-pumps and oil-temperature control are controverted. Five elements necessary for an efficient cooling system are enumerated.

ONLY a small percentage of the total heat value of the fuel-air mixture inducted into an internal-combustion engine is usefully employed. About 30 to 40 per cent is carried off by the exhaust gases, 4 to 5 per cent is lost by direct radiation and, depending upon the design, from 25 to 35 per cent must be eliminated by the cooling system. That the fuel energy dissipated by the cooling system must be wasted is especially unfortunate, since energy must be spent to accomplish this waste. However, mechanical limitations make imperative the devising of a cooling system that will maintain cylinder and combustion-chamber-wall temperatures which will assure satisfactory and continuous operation of the engine.

The two principal methods of cooling internal-combustion engines are directly by air and indirectly by liquid. This paper is concerned mainly with liquid cooling.

To solve the cooling problem for conventionally liquid-cooled internal-combustion engines, two heat-transfer equations must be satisfied; one deals with transfer of heat from the engine surfaces to the liquid cooling medium and the other with its transfer from the liquid cooling medium to the air. In either case, to produce a condition of temperature balance, all of the heat dissipated by the engine must be transferred. Quantitatively, for practical purposes, this can be expressed by

$$H = P \times K \qquad (1)$$

where

H = total heat to be dissipated, in British thermal units

K = heat transferred to cooling liquid, in British thermal units per brake horsepower per minute.

P = brake horsepower developed

The average values of K for water, as shown by tests, are respectively 60 and 70 B.t.u. per hp. per min. for medium-displacement overhead-valve engines and L-head engines. For large engines, these values may be reduced to 45 and 55, respectively.

The total heat carried to the radiator per minute by the cooling liquid is the product of pounds of liquid circulated per minute, temperature rise of the liquid in passing through the engine and specific heat of the liquid. The total heat can easily be determined by metering or weighing the quantity of water circulated through the engine and reading the rise in temperature on a thermometer. Shown as an equation

$$H = W\,(t - t_1)\,C_l \qquad (2)$$

where

C_l = specific heat of the liquid, in British thermal units per pound

t = temperature of liquid leaving engine, in degrees

t_1 = temperature of liquid entering engine, in degrees

W = weight of liquid circulated, in pounds per minute

For water, which has a specific heat of 1, this equation becomes

$$H = W\,(t - t_1) \qquad (3)$$

Equation (3) shows that only the heat input to the cooling liquid and the number of pounds of liquid circulated per minute affect the temperature drop from the top to the bottom of the radiator or the temperature rise through the engine. The fan has no bearing on this factor; nor has the radiator, other than its effect on the flow characteristics of the fluid. If the radiator offers resistance to the flow, reducing the quantity circulated, an increased temperature drop will result. This temperature drop governs the rate of heat convection from the core for a given weight of air passed through the core, assuming like top-tank and air temperatures. Small temperature drops are desirable, because they maintain higher mean temperatures of the radiator surfaces and establish a higher differential between them and the air; hence more heat will be convected. They are particularly advantageous where high air temperatures prevail.

Conditions for Temperature Balance

To maintain a condition of temperature balance, the heat dissipated into the air by the radiator core must be quantitatively equal to the heat dissipated to the cooling fluid by the engine. Expressed as an equation

$$H = W_1 \times C_a \times t_a \qquad (4)$$

where

C_a = specific heat of the air in British thermal units per pound

t_a = temperature rise of air passing through the radiator core, in degrees fahrenheit

W_1 = weight of air raised in temperature, in pounds per minute

Analyzing further, we have

$$H = A \times v \times W_a \times C_a \times t_a \qquad (5)$$

where

A = frontal area of radiator, in square feet

v = average velocity of air through the radiator, in cubic feet per minute.

The specific heat of air is a constant: 0.24 B.t.u. per lb. The specific weight of dry air at 110 deg. fahr. is used to take care of high ambient-air temperatures. It is also a constant, generally taken to be 0.070 lb. per cu. ft.

Reference to Equation (5) discloses three variables: core frontal area, core efficiency and air flow, each being a factor in effecting the total heat transfer.

This groundwork should establish the necessity for

[1] M.S.A.E.—President, chief engineer, Schwitzer-Cummins Co., Indianapolis.

giving full consideration to the cooling problem as a whole, rather than from the standpoint of any particular unit. A correctly developed cooling system must incorporate these three factors in their most economical relation to one another, considering cost, weight and horsepower expended.

A radiator frontal area should be selected that will allow fan sizes and speeds proportional to the permissible power for driving the fan, usually assumed to be about 5 per cent of the total engine output; and attention should be given to the fan-driving means. Radiator-core thickness or resistance to air flow must not be increased to such an extent that, with the fan adopted, the product of pounds of air times its temperature rise will be less than a maximum. This product provides a very satisfactory way of comparing the efficiency of different radiator cores of equal frontal area considered for use with the fan chosen for a given cooling system.

The foregoing three factors represent the three units of which any liquid cooling system is composed; namely, the radiator, as expressed by frontal area and temperature rise; the fan, as expressed by air volume delivered through the radiator; and the liquid circulation, as expressed by the temperature drop of the water through the core, which influences the temperature rise of the air through the core.

Fan Selection and Installation

As air passing through the radiator is the primary means of convecting the heat from the liquid, the fan is the governing factor in cooling; therefore determination of its correct type and installation are of utmost importance. The volume of air delivered should be considered first. The type of fan most widely used is the disc fan, to which class belong all fans that take their air supply mainly from an area immediately behind their peripheral circles and discharge it substantially at right angles to their plane of rotation. They deliver large volumes of air only at very low pressures. The large volume is very valuable, but the low pressure is a decided disadvantage. However, automatic radiators have been designed with this point in view and offer small resistance to air flow.

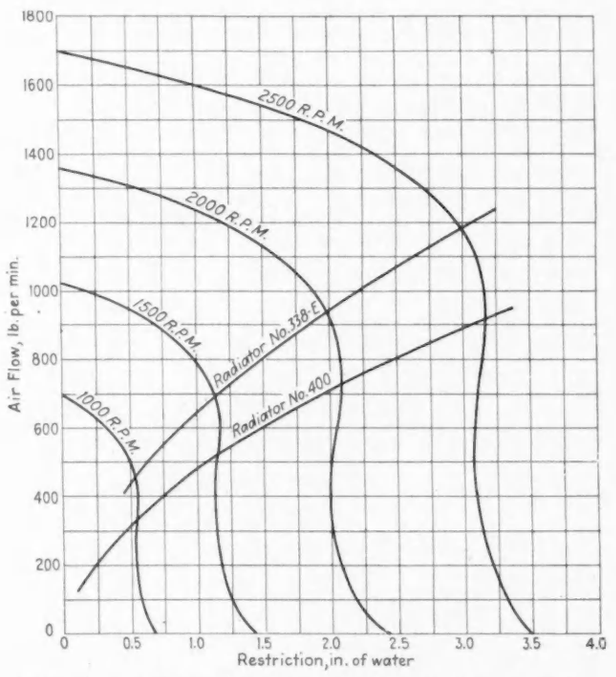


FIG. 1—AIR-FLOW-RESTRICTION CURVES FOR A 26-IN.-DIAMETER TAPER-BLADE FAN AT VARIOUS FAN SPEEDS, AND SIMILAR CURVES FOR RADIATORS NOS. 400 AND 338-E

The Radiator Air-Flows Are Computed for a Radiator-Core Area of 731 Sq. In.

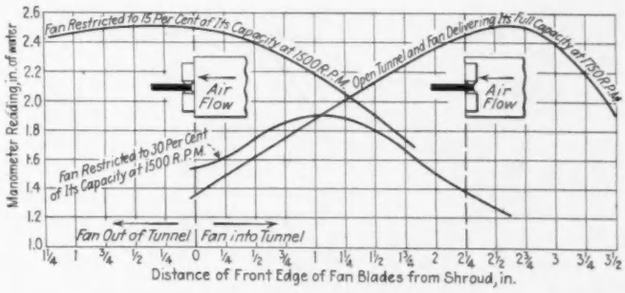

Fan Restricted to 15 Per Cent of Its Capacity at 1500 R.P.M. Open Tunnel and Fan Delivering Its Full Capacity at 1500 R.P.M.
Air Flow
Air Flow
Fan Restricted to 30 Per Cent of Its Capacity at 1500 R.P.M.
Fan Out of Tunnel
Fan into Tunnel
Manometer Reading, in. of water
Distance of Front Edge of Fan Blades from Shroud, in.

FIG. 2—EFFECT OF POSITION OF FAN WITH RELATION TO SHROUD AGAINST DIFFERENT RESISTANCES

The Manometer Reading Is Considered as a Function of Air Flow

Fig. 1 shows the performance characteristics of a 26-in. fan at several speeds, plotted with the same coordinates as those for the static-resistance characteristics of two radiator cores. The points at which the curves cross indicate the air flow that would result from using the fan and the radiators represented by the curves. Radiator No. 400, having the higher resistance characteristics, permits the smaller air flow for a given fan speed. Assuming a constant entering-air temperature, each radiator effects a definite temperature rise for the air flow represented at a given fan speed. The product of the temperature rise and the air flow for a given fan speed is an index by which the radiators can be compared when the fan condition is held constant. Similar indexes for comparisons when the fan speed is varied can be obtained in the same way.

The characteristic performance curve of a normal fan shows that, beyond a certain fan speed, the horsepower requirements are entirely out of proportion to the air delivery. The power required increases as the cube, and the delivery only as the first power, of the fan speed. Therefore large-diameter fans should be used at slow fan speeds. For slow-moving vehicles and stationary installations, fan housings should be employed to assure the maximum efficiency of the fan and also the most uniform air distribution through the core. When static pressure against the fan does not prevail, the housing must be designed so that the fan will be entirely inside of it. If, however, the fan must operate against pressure, the position of the fan with relation to the housing changes accordingly.

Curves in Fig. 2 show that the point of highest efficiency depends upon the static pressure against which the fan must operate. The discharge side of the fan is of importance. Disc-fan characteristics make necessary the provision of free outlets to overcome the handicap of low pressure. The air under the hood expands because of its increased temperature, and inadequate outlets result in the building up of a static pressure that seriously interferes with the fan capacity. Since the occurrence of vapor lock in the fuel supply, free outlets and adequate hood ventilation are recognized as necessary. If possible, the air blast should have an unobstructed passage to the fuel-pump, carbureter and fuel lines. Adequate air movement under the hood will reduce the temperature there and improve vapor-lock conditions.

A fan drive that will assure constant and economical operation should be provided, dimensioned in proportion to the power required to drive the fan. Pulleys should be large, to give sufficient contact and reduce the necessary belt tension. If V-belts of 1-in. width

are inadequate, multiple belts should be used. Two ½-in. belts will give better service than a single 1-in. belt.

Objections to Automatic Fan Clutches

Passenger-car engineers periodically attempt to reduce fan horsepower and noise or gain better thermostatic control by incorporating a clutch in the fan drive. These attempts are made along three lines: (*a*) a clutch in the fan drive that will slip when a predetermined fan speed is reached, the speed being held approximately constant thereafter; (*b*) a clutch that will completely disengage at a predetermined fan speed, the fan being no longer driven, and (*c*) a fan that will be driven through a thermostatically controlled clutch only when the engine has reached the right operating temperature. The idea of using either the slipping clutch or the throw-out clutch usually is based upon the erroneous impression that the fan is not required for high-car-speed running. Repeated high-speed tests show, however, that the fan contributes a reduction of 13 to 20 deg. in top-tank temperature. No satisfactory and feasible design of slipping clutch has been evolved that will dissipate the heat developed by the increase of constant-torque power beyond the point where slip begins. All devices of this type that have been used have been discontinued because they burned out.

The greatest objection to throw-out clutches is that they draw attention to fan noise each time the device picks up or lets go of the fan. On devices we have made, we found it necessary to provide a soft engagement so that, when the device picked up the fan, damaging shock would be avoided. But with such an engagement, the slipping-clutch problem presents itself if the car is driven constantly in the range of pick-up speeds. We have found much the same objection to a thermostatically controlled fan. At certain air temperatures attention is periodically drawn to the fan as the device engages or disengages. This device also would be subject to the slipping-clutch objection if driven continuously in pick-up speeds.

Experience in designing and building various types of clutching device into fans indicates that no satisfactory design can be made without considerable complication and expense, which might better be employed in the direction of improving water-pump, radiator and fan efficiency.

Uneven Fan-Blade Spacing Not Advantageous

Fan designs in which the blades are spaced at unequal intervals have been adopted by several automobile manufacturers. The theory is that the frequency of the vibrations can be varied so as to produce interfering sound waves that damp or counteract each other, thereby reducing the magnitude of the emitted sound. We have experimented on a dozen or more different automobile models and have yet to find a worthwhile advantage in using uneven blade spacing. Some engineers are under the impression that air flow is increased by uneven spacing, but repeated checks show this to be incorrect.

Fig. 3 shows that the air-delivery of a four-blade fan falls off directly as the blade spacing is reduced from 90 deg., while the horsepower requirement remains entirely normal. Fig. 4 gives this same information for a six-blade fan as the blade spacing is varied from 60 to 0 deg., the air delivery falling off directly with the reduction in spacing. For a spacing of 30 deg., the delivery is reduced about 10 per cent from that at the normal 60-deg. spacing, and the power by the same amount. The characteristics of the same six-blade fan with equal blade spacing, and with air delivery and horsepower plotted against fan speed, are shown by Fig. 5. For a 10-per-cent reduction in air flow at 2400 r.p.m., a reduction of about 23 per cent in horsepower can be realized.

Quieter operation would result at this air delivery, as fan noise and horsepower are closely correlated. We have found that, if the delivery from a fan of unequal blade spacing is equalled with a fan of approximately the same design but with blades evenly spaced, the noise characteristics in most cases will be similar.

Results of Acoustical Tests

The results of acoustical tests sum up about as follows: The frequency of the sound emitted by a fan having evenly spaced blades is the product of the speed and the number of blades; if the spacing is uneven or in groups, two frequencies will result, one equal to the speed times the number of blades and the other equal to the speed times the number of groups. For example, if two blades in a six-blade fan are alternately close together so as to form three widely spaced groups, two frequencies will be emitted, one six times the speed and the other three times the speed. The predominating frequency would depend upon the spacing; thus, if the blades in each group were very close together, the group pitch would be more noticeable; if the blades in each group were spaced at a considerable distance, the blade pitch would predominate. If a fan should have 20 or more blades with no two spaces equal, the result probably would be a very complicated noise. Should the blades be arranged so that the frequencies of the com-

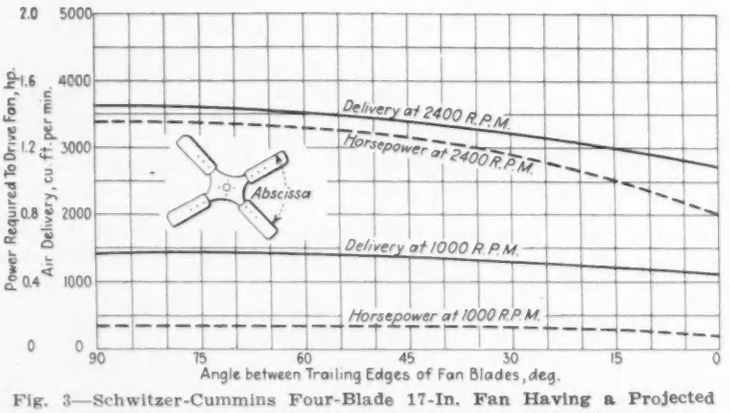


Fig. 3—Schwitzer-Cummins Four-Blade 17-In. Fan Having a Projected Width of 1⅞ In., with Included Angles between the Blades Varying from 90 to 0 Deg.

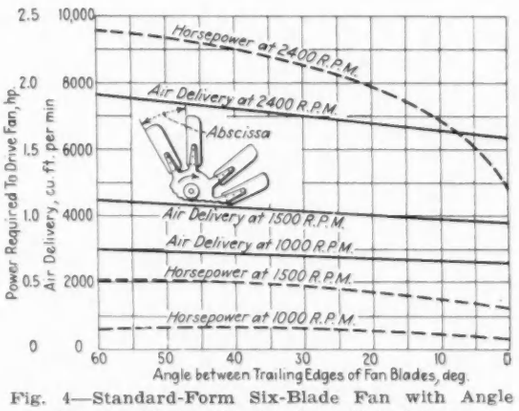


Fig. 4—Standard-Form Six-Blade Fan with Angle between Blades Varying from 60 to 0 Deg.

AIR-DELIVERY AND HORSEPOWER CHARACTERISTICS OF FANS HAVING UNEVENLY SPACED BLADES

ponent vibrations could have no common denominator, the result might be desirable. However, as we must use fans having a small number of blades, because of the higher efficiency and lower cost, the blade spacing does not seem to make a major difference.

Free Liquid Circulation Is Important

Forced or pump circulation of the cooling liquid is a virtual necessity in modern multi-cylinder engines, because of their length, high speed and high compression. The selection of a desirable temperature drop through the radiator core, not to exceed 10 deg., and the known number of heat units to be carried into the core determine the quantity of liquid to be circulated. The formula is $H = W \times (t - t_1)$.

Unfortunately, because of their design, the frictional resistance of radiators, particularly those designed for automobiles and high-speed trucks, limits the water circulation. A compromise pump should be designed that will furnish maximum circulation up to an engine speed of about 2500 r.p.m. and then level off the de-

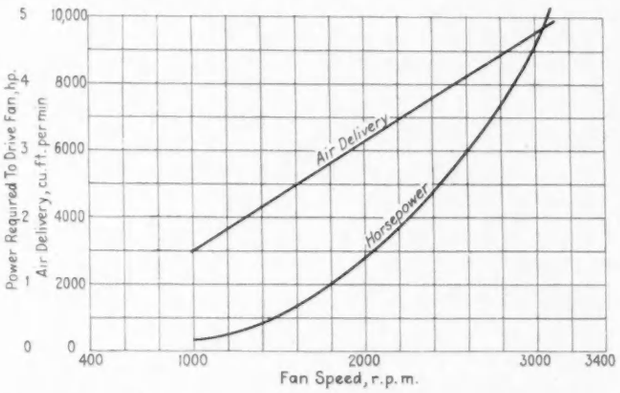


FIG. 5—AIR-DELIVERY AND HORSEPOWER CHARACTERISTICS OF STANDARD-FORM 19-IN. FAN HAVING A PROJECTED WIDTH OF BLADE OF 1¾ IN.

livery to the flow capacity of the radiator. If the radiator core should be unable to return the water to the pump inlet fast enough, a high vacuum will form in the pump-inlet line and the reduced pressure will lower the boiling point in this line, resulting in the formation of steam and producing a tendency toward cavitation. Vapor lock will then occur.

Prevention of cavitation is therefore an important consideration in a pump circulating system. Reduction of the suction head against which the pump must operate, by providing sufficient flow in the radiator and large inlet pipes to the water-pump, is a step toward preventing cavitation; but, in addition, a pressure head must be maintained against the pump discharge at top engine-speeds. A maximum water pressure in the cylinder block of 4 to 6 lb. per sq. in. at a top-tank temperature of 200 deg. fahr. usually will prevent cavitation, and avoid cooling difficulties arising from this source. However, the pressure in the system should not be higher than is necessary to avoid the trouble, as high pressure reduces water flow, which obviously is objectionable.

Water-pump designs have been improved in the last few years but the leaking of pumps still causes many service complaints. The conventional centrifugal pump has given way to a type that is more correctly named "peripheral," in which the water enters at the side near the periphery and is discharged tangentially (See Fig. 6). Pumps of this type have shown considerably increased capacity as compared with other pumps of the same size. More attention is being given to the de-

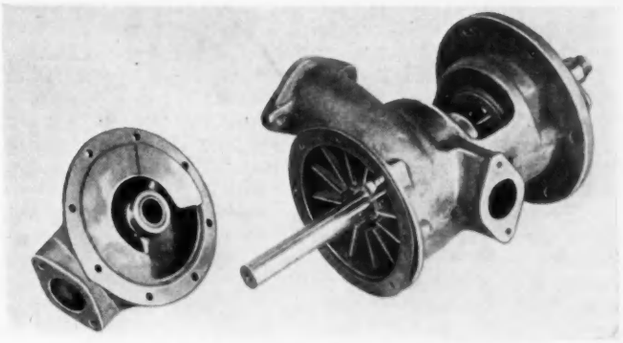


FIG. 6—PERIPHERAL TYPE OF WATER-PUMP
The Water Enters at the Side, Near the Periphery, and Is Discharged Tangentially

sign of the bearing and packing arrangements. Lately, automatic seals have been developed that entirely eliminate the packing. An interesting design is shown in Fig. 7. To maintain the sealing surface, the shaft is mounted on ball bearings and the shaft and bearings are completely insulated from the water, so that no damaging corrosion can occur. This design has the further advantage of preventing grease from being introduced into the cooling liquid. Grease interferes seriously with the rate of heat transfer from the cylinder jackets and radiator core.

Trend to Side Location of Pump

Front mounting of pumps came into favor some years ago, but recently the trend has been back to pumps mounted at the side. Good distribution of the water is more difficult with pumps mounted at the front, particularly in long engines, and if the construction is satisfactorily built the front-mounted pump with drive from the fan belt is as costly as the side-mounted pump with separate drive.

Impellers of small diameter are required in head-mounted pumps because of lack of available space, particularly when the impeller is placed inside the cylinder block. This calls for high pump-speed to assure adequate circulation, especially at low engine-speeds. High pump-speed results in a poor fan condition, for, to avoid noise, fans of small diameter are used. The horsepower requirement per unit air delivery varies almost inversely as the square of the fan diameter; therefore the reduced efficiency of the small fan adds to the already excessive combined pulley load of the normal fan and the

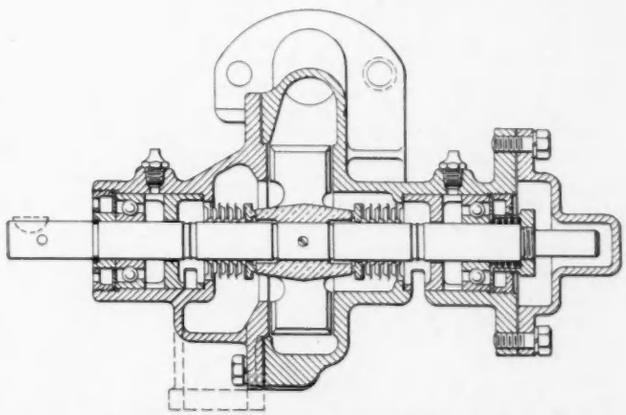


FIG. 7—CROSS-SECTION OF PUMP SHOWING AUTOMATIC SEAL OF BEARINGS, MAKING PACKING UNNECESSARY

water-pump. Further, the fan center is forced low in most head-mounted pumps and the fan must discharge against the engine cross-member of the chain case.

Coupled with all these points is the fact that head-mounted pumps often give mechanical trouble, such as leaking, bearing failure and fan-blade breakage; also, where an adjustable pulley is used on the pump, change in adjustment changes the drive ratio and a "beat" or synchronous vibration often results.

The paradox of this design is that almost invariably the reason for its adoption is its apparent cost advantage through the elimination of either the fan or the pump-unit expense. Actually, a pump and separate fan, both designed to be best adapted for their purposes, can be made at no greater cost than a well-designed head-mounted pump. Such an arrangement results in a better water distribution, reduced fan horsepower and noise and increased radiator efficiency, which is reflected in the cost of the core.

Considerations in Water-Jacket Design

The function of the water-pump is to discharge into the water-jacket a sufficient volume of water to carry from the cylinder walls the heat conducted through these walls. It must be capable of doing that; otherwise flash points will occur. If the heat input, the mechanical work to be done and the heat units in excess of the work requirements are known, the designing of a water-jacket apparently should be simple. The volume that a water-jacket must contain does not give the answer to the problem; distribution of the water in the block is of much greater importance. Modern engineering practice places the deepest layer of cooling liquid over the top of the cylinders, where it should be; but foundry practice and the rigidity required have interposed ribs and columns which act as retarders and, if the water spaces are not interconnected with large openings, steam pockets will be formed. These will reveal their presence by sudden increases of top temperatures, particularly when the engine stops.

The layer of liquid should be arranged so that the circulation system will allow the entire volume of liquid to flow at a uniform rate. The cold water is directed first to the valves. If the water inlet is on the opposite side of the block, the distributor plate, if one is used, may be provided with extensions between the cylinders to discharge the cooler water directly around the valves. In one case of an overhead-valve engine, in which difficulty from valve warpage was experienced, the cool water was discharged directly into the cylinder head and the block was cooled by gravity circulation. This eliminated the trouble.

A clean water-space free from grease is necessary for good heat transfer. Tests have been made of a firebox in which the hot gases impinged on one side and the other side was in clean contact with water at a temperature of 212 deg. fahr. The temperature of the gas side was only 40 deg. fahr. higher than the water temperature. When a greasy deposit was present, the temperature difference rose to 550 deg. fahr. Therefore, frequent flushing of the cooling system is necessary.

The gas film in the combustion chamber offers much higher resistance to the heat flow than does the oil film; in fact, it represents 95 per cent of the total resistance. The temperature gradients that must exist so that the heat will be rapidly conducted from the explosion side of the cylinder to the water side give rise to rather high temperature differences between various parts of the metal and to high local temperatures at such places as the center of the pistons and at the valves. These temperature differences cause unequal expansion and corresponding stresses, and result in a tendency toward preignition. Factors that determine the temperature of the metal at any point on the inner surface of the walls are the mean rate of heat flow into the wall per unit area at that point and the thickness of the wall at the same point. The real heat flow will depend upon the density, the temperature and particularly the turbulence of the gas. This is one of the reasons why some engines cool more easily than do others.

Controlling Oil Temperature by Jacket Design

Maintenance of suitable oil temperatures is another problem of engine cooling. In general, we have found by extensive tests, particularly at high-speed full-open-throttle operation, that oil temperatures must be kept below 250 deg. fahr. when the air temperature is 100 deg. fahr.

We feel that the general belief that oil is cooled almost entirely by radiation through the oil-pan is erroneous. We have found that oil temperatures can be controlled to a great extent by design of the water-jackets. The more contact the oil-wetted surface has with a wall that is water-wetted on the opposite side, the lower will be the final oil temperature. This indicates the desirability of surrounding the cylinder barrels with water for their full length. Engines so constructed have been found to be capable of operating continuously at top speed without the use of auxiliary oil-coolers, maintaining oil temperatures within 30 deg. of top-tank water temperatures. Where, because of design reasons, full-length water-jacketing is not feasible, the use of oil coolers is a valuable asset.

Mechanical Details of Radiators

The radiator industry is now concentrating on three designs: the fin-and-tube, the cellular and the cartridge types. The manufacturers seem to attain about the same efficiency for the three types, but the first two are the most popular. Although the designs are fundamentally the same, certain construction features differ; however, these eventually lead to the use of radiating material of approximately equal efficiency. In planning the radiator, the usual procedure is for the car architect to establish the shape, the purchasing agent to fix the cost limit, and the design engineer may go as far as he chooses within these restrictions.

Knowing the characteristics of the core with respect to static resistance and temperature rise of air passing through it, the correct radiator can be selected by use of the heat-balance formula, Equations (4) and (5).

In a problem involving as many variables as does the cooling system, it is desirable at least that radiator cores be manufactured as carefully as possible. Adequate water spaces, correct bonding and keeping the air passages free from solder obstructions are factors that affect the efficiency.

The mechanical performance of the radiator is dependent primarily upon the design. Radiators should be capable of withstanding much vibration and shaking without disintegrating. Supports and mountings should prevent flexing through the core area. A decided increase generally is needed in the size of the upper tanks, which in general are often too small for high engine-speeds as a considerable volume of air unavoidably enters the circulating water either by natural absorption or through water-pump stuffing-boxes. When the temperature rises, this air expands and drives out the small quantity of water that conventional small top-tanks contain and, when the top of the core is no longer submerged, distribution of water through the core is upset and a considerable reduction of efficiency results. We have found baffles are very poorly designed for securing the best distribution through the core. Correct

construction of baffles usually prevents the spewing of water from the radiator when the engine is suddenly accelerated or the car is quickly decelerated.

Suitable lower-tank construction has sometimes been neglected. The lower tank should be deep enough to permit adequate return of the water to the pump-inlet line. In some cases holes through the lower tank for engaging the hand crank have seriously interfered with circulation. Extra capacity should be provided to take care of sludge, grease and other matter that tends to collect in the lower tank. Outlets should have as large an area as possible to prevent short-circuiting of the water through the core and to reduce the resistance to flow. Oblong openings are highly desirable.

A few years ago radiators that had a tendency to be clogged on the front and in the cells by insects and dirt were condemned, but seemingly this is not a serious matter today, as fin spacing is as close as eight or nine per inch in fin-and-tube radiators. In certain industrial applications such spacing doubtless would develop trouble from clogging; so, when selecting a radiator, it would be well to bear in mind the nature of the service.

Shutters and grids have added greatly to the problem of correct air flow and, by poor location relative to the core, have been responsible for the creation of noise.

Summarizing, the elements necessary for an efficient cooling system are:

(1) An efficient fan well designed in all its details
(2) Provisions for air outlets
(3) An adequate fan drive
(4) Radiator and water-jacket design that will allow a cooling-liquid circulation adequate to maintain reasonably small temperature drops and provide suitable distribution in the system
(5) A radiator core which, when used with the fan selected, will pass a weight of air sufficiently raised in temperature that the product of pounds of air and temperature rise is a maximum

Alloys-of-Iron Research

SUBSTANTIAL progress in both phases of its work during the last year is reported by the Iron-Alloys Research Committee of the Engineering Foundation, on which the S.A.E. is represented by T. H. Wickenden. The critical survey of research on iron and its alloys is reported as more than 50 per cent complete, and the publication program is well advanced. The manuscript for the first of a series of 10 monographs has been delivered to the publisher and several others are in various stages of preparation.

Of the 2000 journals in 10 or more languages which are likely to contain articles of interest in this program, approximately 150 have editorial content that is confined to reports of metallurgical research or to articles of marked practical value. The review has been confined to articles in the most important metallurgical journals, more than one-half of which, in English, German, French, Swedish and Italian, have been abstracted and information for monographs and manuals filed. An aggregate of 3540 technical and scientific papers have been reviewed and 9419 abstracts prepared.

The detailed classification of abstracts now in the file, as indicated in the accompanying table, is fairly representative of the information available in the literature on ferrous metallurgical research, but the proportion of abstracts on manufacture, heat-treatment and uses probably will increase substantially as more trade journals are reviewed.

The literature survey eventually will make available a critical summary of all important papers recording research on ferrous metals published since 1890, and the information available will make possible the compilation of a comprehensive bibliography on any phase of ferrous metallurgy.

The monographs at present planned are: 1, Iron-Molybdenum; 2, Iron-Silicon; 3 and 4, Iron-Tungsten and Iron-Copper; 5, Pure Iron; 6, Iron-Nickel; 7, 8, 9 and 10, Iron-Chromium, Iron-Vanadium, Iron-Manganese and Iron-Carbon Alloys.

TABLE 1—CLASSIFIED ABSTRACTS OF ALLOYS OF IRON WITH 39 ELEMENTS[a]

	Sulphur	Phosphorus	Manganese	Silicon	Nickel	Nickel-Chromium	Chromium	Vanadium	Molybdenum	Tungsten	Aluminum	Copper	Cobalt	Zirconium	Uranium	Titanium	Arsenic	Boron	Cerium and Beryllium	Zinc	Tin	Lead, Bismuth and Antimony	Alkali Metals	Rare Metals	Oxygen	Hydrogen	Nitrogen	Carbon Monoxide	Carbon Dioxide	Other Gases	Oxides	Sulphides	Silicates	Total
Historical		2	15	12	6	2	9	3	5	11	7	3				6	1			1	1			1	2	1	3	1					1	93
Mining and Smelting	5	7	38	97	13	1	27	17	15	16	3		9	3	4	19	2	3	6	1		1	1	2	2	1		1						294
Manufacture																																		
Melting, Refining	114	44	78	114	25	14	39	14	12	13	30	11		11	5	30	2	1	3	2	2	4	13	2	22	9	19	15	4	1	28	6	14	701
Ingots, Castings	37	26	39	83	21	15	15	6	5	4	13	2				15	1	2			1				9	12	11	16	5	5	3	2	7	355
Hot Working	14	11	20	33	23	12	25	9	11	22	6	20	2	4		2	8	2			5	8		2	12	1	3	1			6	2	2	266
Cold Working	2	3	5	7	9	4	12		1		1														2	1	1				2	1	1	52
Constitution																																		
Miscellaneous	24	20	28	61	18	4	24	13	9	25	10	3	3	3	1	11	1	3		7	4	2	4	8	32	38	50	28	14	4	11	4	8	475
Phase Diagrams	36	33	62	150	105	14	93	18	36	54	26	30	35	1	2	11	7	14	7	13	13	21	4	14	30	12	27	9	5	1	5	3		891
Structure	40	57	72	154	83	15	84	20	29	52	23	13	9	7	4	20	4	7	3	15	7	2		4	30	7	45	1	2	1	35	26	20	891
Microstructure	9	16	8	8	23	11	18	2	1	5	4	2				1	3								1	3	2	3		1	12	2	1	136
Heat-Treatment	14	21	33	83	34	17	63	12	23	35	13	4	10	1		6		1		2	1	1			3	6	17	1	2	1	6	3	2	415
Properties																																		
Constants	4	5	30	70	43	12	30	9	8	19	8	3	12	1		6			2	1		1		2	4	2	5				1	1	1	280
Mechanical	50	61	125	236	164	65	137	69	67	39	26	34	13	10	9	43	9	9	7	5	9	2	3	3	17	9	31	4	2	1	2	4	12	1,277
Magnetic	1	4	31	117	51	7	31	3	9	35	13	6	24			4	4				2	2			5	1	6	1						357
Electrical	1	6	24	50	35	11	28	6	11	13	12	5	6			2	2				1	7		5	1	2	3	1	1					233
Corrosion	14	12	34	112	75	24	108	14	25	29	19	63	14	3	1	5	3	3	2	12	1	4		4	7	4	7	1	1		3	2	5	611
Miscellaneous	13	15	32	54	40	10	57	25	23	52	5	8	17	1	3	2	8		1	1	8	4		4	3		8	1	1	1	1		1	399
Effect of Heat-Treatment	5	3	56	45	80	35	103	41	41	56	7	10	11	3	2	2				1				1	3		10				1		1	517
Effect of Temperature	5	6	22	33	59	22	69	22	31	40	8	9	11	1	2	2		1			1			1	1		1	1						348
Heat-Treatment	17	9	33	55	30	20	57	20	18	49	18	8	10	3	1	7	2	2	1	4	1	1		1	11	2	43	8	8	3	10	2		454
Uses	3	7	40	70	46	33	62	17	27	26	9	12	9		1	2		1				1		2			3			1	1	2		374
Total	408	368	825	1,644	983	348	1,091	340	407	595	261	246	195	52	36	198	54	49	32	65	57	61	25	55	197	111	295	93	45	20	127	60	76	9,419

[a] Oxides, sulphides, silicates, carbon monoxide, carbon dioxide and so forth are here considered as elements.

Free-Wheeling Devices and Their Control

Discussion on Austin M. Wolf's Semi-Annual Meeting Paper[1]

S. O. WHITE[2]:—Unquestionably Mr. Wolf's paper represents considerable effort on his part and brings together more concrete information as to free-wheeling design and experience than I have seen in any one contribution.

As the paper discloses, the problem involved much more than any of us supposed when we first took up the idea of applying a one-way clutch to an automobile transmission. Considerable work had already been done abroad, and a study of the information available in the United States Patent Office revealed many different ways of providing a one-way clutch. Many kinds of such clutch were in successful operation in varied applications on machinery. The fact soon became apparent, however, that satisfactory operation on a machinery installation was no assurance that the same device would be satisfactory in an automobile transmission.

Evidently the idea is current among laymen and even engineers that a one-way clutch that will operate successfully as a hand-made bench model or in some light car will serve for all automobiles, and we are constantly besieged with one-way clutch suggestions. Most of these are slight modifications of old ideas, but we must look at all of them, as possibly someone might devise a system that would be much better than any we know of now. We are not now aware, however, of any device that is more generally satisfactory and foolproof, from both the manufacturing and the operation viewpoints, than is the roller clutch.

One of the earliest clutches we tried out was of the screw-and-nut type, somewhat related to the well-known bicycle coaster brake, but this would engage with a thump and did not give a very good account of itself.

Numerous one-way clutches employ friction shoes actuated by cams, rollers, wedges, springs and the like, but we have not found any of these that would be satisfactory. In some cases the shoes are held out by centrifugal force so as to create friction, and usually the shoes will not take hold quickly enough because of the oil film that must be squeezed out from between the engaging surfaces, or they tend to slip or take hold with a jerk, do not let go promptly enough or are too expensive to manufacture.

There are various ratchet types other than roller ratchets, and a considerable class which we call the "strut type," embodying various applications of the strut principle, which are free in one direction but compress the strut against movement in the opposite direction.

Some devices could perhaps be made to operate satisfactorily if they were given the amount of concentrated effort that has been given to the roller clutch, but we do not see any prospect of reducing their cost enough to compete with the roller clutch.

The objection has frequently been raised by the proponents of other kinds of one-way clutch devices that roller clutches require extreme accuracy and therefore must be expensive. This accuracy becomes merely a matter of correct tooling and methods, combined with a certain amount of experience and skill. The accuracies necessary in the roller clutch are of no higher order than that required of a high-grade roller bearing.

[1] See S.A.E. JOURNAL, July 1932, p. 265. The author is a consulting engineer in New York City and a Member of the Society.

[2] M.S.A.E.—Chief engineer, Warner Gear Co., Muncie, Ind.

Higher Air Speed at Less Cost

SPEED is the pedestal upon which American business and constructive thought are founded. It is perhaps more true of the aviation industry than of any other enterprise. Even now a trip from the Atlantic to the Pacific Coast made entirely by rail requires 85 hr., as compared with 24 hr. in a regular transport airliner; and plans are being consummated to cut this time to 12 hr. on regularly operated schedules.

The value of the airplane is almost entirely controlled by the speed it makes. Perfect streamlining, incorporated on a correct aerodynamic structure, has allowed us to take the engine and pilot from one airplane that will cruise at 100 m.p.h., and install them in a modern plane that will cruise at 150 m.p.h., and, with identical loads and the same engine, prove that the high-speed plane is much less expensive to operate.

Manufacturers of high-speed planes have been greatly handicapped in comparison with manufacturers of conventional-speed airplanes by reason of lack of data that are pertinent to the higher stresses which ensue with greater speeds; however, one of the tremendously interesting facts about the modern high-speed transport is that it embodies all of the characteristics of the high-speed airplane with the slow-landing ability of its predecessors.

The cost of operating the most recently developed air transport with this same engine is approximately $25 per hr. for a distance of 175 miles as against the same cost for 100 miles for another type of equipment, equally as expensive in its original purchase. This is a saving of over 40 per cent for every mile operated.

Operators who have had the foresight necessary to make money out of aerial transportation have operated, since March, 1931, modern transport planes at consistent cruising speeds of 175 m.p.h. on regularly operated passenger schedules. The Varney Air Service is operating on a schedule of 1 hr. 8 min. from Los Angeles to San Francisco, fully loaded, with equipment able to cruise better than 3 m.p.min., and with a top speed of 220 m.p.h.

Figures show that a trimotor plane using three Wasp engines will fly this same route from Los Angeles to San Francisco in 3½ hr., or 10½ hr. of engine time, with 12 passengers. Putting the same amount of engine time on a modern single-engine transport plane would be the equivalent of five trips with 30 available passenger seats; or 2½ times the number of passenger seats are made available with the same engine time.

The highly efficient single-engine type of monoplane has proved itself capable of cutting by more than one-half the operating cost of an airline running multi-engine designs, or of more than doubling the operating capacity at no increase of equipment or maintenance cost, and it is definitely proved that passengers will ride with equal enthusiasm on a single-engine transport over regular scheduled airlines.

An outstanding airline operator in the Middle West pays his pilots $400 a month, and they fly on an average of approximately 100 hr. per month on a regular schedule calling for 175 m.p.h., or 17,500 miles per pilot per month, which is considerably less cost per pilot than the same operator pays his bus drivers.

An increase in speed of 30 m.p.h. has been gained in the last year on regular transport planes by the use of the retractable landing-gear. A clean low-wing design using this mechanism not only attains terrific top speed but also actually has a slower landing-speed than a high-wing standard monoplane.—Southern California Section paper by Carl B. Squier, manager, Lockheed Aircraft Co., Burbank, Calif.

How the Design Engineer Views Manufacturing

Production Meeting Paper

By Alex Taub[1]

As engineering standards have risen, the need for production ingenuity has become greater than ever before. The engineer looks to the shop for major assistance in realizing his ideals of improved products. He expects the shop voluntarily to reduce the variations from dimensional specifications and to improve its facility to meet changes in design.

Refinement in design is useless unless the shop can accurately hold the dimensions. Powerplant characteristics are largely controlled by the accuracy of centers and roundness and straightness of bores in cylinders and bearings. Crankshaft balance, quiet valve tappets and uniformity of weight and fit of reciprocating parts are all dependent upon accuracy of machine operations.

To be able to make design changes in the product without great expense is vitally important. Tools must be designed with facility for change. Fixed-center boring machines are to blame for considerable engine trouble and may make design changes prohibitively expensive. The machine-tool specialist should be called into consultation while new designs are still in the liquid state. The mental attitude of the men in the shop toward changes in machining methods should be altered, and intimate contact of the shop man with the engineer is needed.

THAT the production specialist's day is over, as has been said, is certainly not true. We believe that the need for production ingenuity is greater today than ever before. Engineering standards have gone up, perhaps because business is down.

The engineer looks to the shop for major assistance in the realization of his improved ideals. He expects from the shop progress along two distinct lines:

(1) Voluntary reduction in variations
(2) Increased mobility for change

By (1) is meant a reduction in variation from the mean specifications, such as usually is represented by tolerances, specified and otherwise.

A group of trouble-makers can be found in structural variations. These can best be illustrated by considering a crankcase and a cylinder block. The walls of these units vary in thickness approximately 3/64 in. If this allowance is included in the specifications as a tolerance, it usually is all plus, which means weight, and somebody must pay for the iron. If this variation is not allowed for in the specifications, we must deal with a possible weakness. This weakness may produce resonance and roughness, and this is not in keeping with current engineering requirements.

A structure like a crankcase is certain to incorporate critical locations which are supersensitive to metal variations. A good crankcase is carefully developed or, we might say, calibrated as is a carbureter, to obtain the maximum structural effect for the minimum weight.

To go through a crankcase with a fine-tooth comb is useless unless some assurance can be had from the shop that the dimensions can be held. When we bear in mind the fact that it is possible to effect a structural gain of 40 per cent in a crankcase by an addition of 3 per cent in weight, the importance of variation can readily be seen.

These variations include those of the foundry and the machine-shop. The machine-shop can help by accurate location points and the rapid check-up of castings as they are received. This will avoid the piling up of unsatisfactory stock, which usually is absorbed in production.

Machine-Shop Variation from Specifications

The characteristics of a powerplant are largely controlled by the accuracy of the centers, roundness and straightness of cylinder and crankshaft and camshaft bores. Considerable progress has been made along these lines. However, if the most is to be made of every square inch of bearing area, more attention must be given to the variations at these places. Engineering decisions are based upon tests of various kinds, and, if the manufacturing variations are not discounted, trouble may follow. If the variations are large, the effect of the investigation is lost.

Another important variation giving considerable field trouble is that from true concentricity of valve-stem guide and bosses with the valve-seat throat. Eccentricity here results in poor valve life and generally an engine performance that is below par. I believe that opportunity exists for development of tools for machining these parts. Improvement in accuracy would be as far-reaching in its satisfaction as eccentricity has been annoying in the field and to the engineer.

Machinists in the shop appreciate quiet tappets, and, although concentric valve seats are not a cure-all for noise, the development of a good, quieting ramp on the camshaft is impossible if the valves are "acting up" on their seats. Here is an opportunity for the machine-tool maker to point the way with new tools or methods for more concentric valve seats. This means quieter, smoother engines and maximum power for a longer operating period. It presents an outstanding opportunity for more result per dollar.

Crankshaft Variation Causes Engineer Concern

Variations in the crankshaft are sources of concern to the engineer. The crankshaft is literally infested with tolerances to cover manufacturing variations and still more that are not listed. No doubt the specified variations are necessary today; but what improvement may we expect? Are you doing anything about straight journals on crankpins and main bearings? What about smoothness of finish, and what is being done to reduce the required tolerance on diameter? What is being done to narrow down variations here in crankpin index and radius?

This last point is today the outstanding element in heavy corrections for out-of-balance conditions. Sometimes we note heavy grinding on cheeks that is all out of proportion to forging variation. This takes time and may weaken the shaft structurally, and, because of the time involved, tends to create a shop demand for wide tolerance for balance. With proper cooperation

[1] M.S.A.E.—Development engineer, Chevrolet Motor Co., Detroit.

between engineer and forge plant, arrangements can be made whereby the crankshaft forging will be ideal for balancing. However, no forging compensation can be made for position error in crankpins. This presents another opportunity for the machine-tool man to show what he can do.

A better understanding of crankshaft balance is required. Today crankshafts are being held to ½ in-oz. out of balance, which is very good; but the important point is, where is the out of balance? The shaft, though checking to ½ in-oz. out of balance, may actually be 24 times this much under operation if consideration is not given to the position of the out of balance and a couple is permitted to exist. Increased tolerance for out of balance is preferred to incorrect position.

The engineer and the forge shop can collaborate to the extent that all forgings shall be heavy in the desired plane. This would still mean nothing if crankpin-position error is permitted. Very little position error of the pins can create considerable out-of-balance effect.

Reciprocating parts of an engine are respected by both the shop and the engineer but, like the crankshaft, they are smothered in tolerances, with resulting weight variations. Selective assembly is the present way out. However, what is being done to bring all rods and pistons to one weight? We need automatic weighing equipment that is easy to use and that will indicate how much must be taken from each piston to bring all to the same weight. This will leave us with only the fit variation for selection. Likewise, connecting-rods should be designed and manufactured to a uniform weight, so that any rod will fit any engine in the field and shop. This means maintaining consideration of interchangeable fits. Rods have been made uniform in the matter of fit; now let us have uniform weight.

We could go through every piece of the car and raise questions as to what is being done to reduce tolerances. For instance, spline shafts are an endless source of trouble because of the tolerances required under present methods of fabrication. What shall be done about it? To ask is easy, but to execute no doubt is difficult.

Need of Mobility for Change

To change with the trend is second in importance only to creating the trend. To do either we must be able to make changes without the tremendous expense of the immediate past. The greatest obstacle to mobility to change is expensive tools. I do not mean by this that high-grade machine-tools cannot be used; there can be no return to the file and solder iron. I mean that both large and small tools must be designed with facility for change.

One of the worst examples of immobile tools is cylinder-boring equipment provided with fixed heads. In the last few years, as you know, the reamer has been applied to the engine of our industry, and in most cases the larger bore was obtained without any change in bore centers. Reduction in water-passage area was the result, with its many evils for foundry and field. Uneven cooling of bores, which follows such restrictions, will give trouble through the life of the car. It breeds blow-by and hot oil, compression leaks and power loss, worn piston-rings and excessive oil consumption. Such an engine may be bigger, but it certainly is not better. However, the *centers have been saved.* Fixed-center machinery is to blame.

We must have rugged accuracy without fixed centers. This principle applies throughout. Progress may demand the shifting of one hole in a group. Fixed-center machinery may make the change prohibitive or at least costly, hence a product that is unsatisfactory at some point is continued through production because of the cost of changing. This does not mean that we must go back to the single-spindle machine; it calls for equipment with adjustable centers without sacrifice of accuracy.

Location holes for positioning the product in the jigs are necessary, but suppose that these dowel holes are in such a location that a needed change would require that they be moved a little. Today, in some shops, that would be a catastrophe; it would affect every jig in the cylinder-block line. Why should this be so? Location dowels should be incorporated in fixtures in such a way that they could be moved or firmly fixed at will in the shop.

Shop Reluctance to Change of Method

The mental attitude of the men in the shop to change in machining methods to facilitate changes in the product is worthy of mention. Suppose it becomes necessary to increase the rigidity of a crankshaft, which may require cheeking. Considerable protest arises in a shop that is not cheeking the crankshaft, which is a difficult and unsatisfactory operation. Why? Probably because suitable tools are not available or the sequence of operations would be disturbed. The shop is entitled to its "rathers," but shop habits also must undergo development in mobility to change.

Good engineering ideas or principles often are doomed because the designs cannot be made economically if at all. But this is the day of new ideas; if competition is the life of trade, new ideas are the life blood of competition.

The suggestion has been made that the engineer call in the machine-tool specialist to collaborate on designs while they are still liquid. This might lead to the successful production of an item heretofore regarded as beyond redemption.

What would our associates in the production department do if the engineer told them how to do their job or where they should buy machine-tools? Yet new ideas must be given consideration even when apparently impractical from the producing standpoint. The outside specialist, with his incentive for new-tool sales, should be brought into consultation, but by the shop. Rather than discard a new item, the shop should pass the problem of its fabrication out among the various tool companies in which it has confidence. Thus, casting the bread of our ideas upon the waters of machine-tool genius may result in a return other than just wet bread.

Intimate Engineering and Shop Contact Vital

We ask for a *voluntary decrease in variations* and *improved mobility to change.* These must not be conflicting requirements. One without the other would be insufficient. Appreciation of each of these elements exists today. However, they need more consideration, particularly when appropriations for new or changed products are in the offing. If new investments are in order, let us give consideration to these principles. Who knows but that the promotion of these very principles may help the appropriation along.

The fact has become increasingly evident that intimate contact between the shop and the engineering department is needed to build the proper appreciation of each other's problems. It is essential that the engineer shall know his shop and shop man, so as to temper his suggestions to prevailing conditions. It is just as vital that the shop man be intimate with engineering test results, so that he can see the parts he is producing in the light of their functioning. Test results, or even seeing tests conducted, may influence valuable suggestions as to improved design that will be *easily produced good.*

Six-Wheel Trucks

Transportation Meeting Paper

By Austin M. Wolf[1]

SIX-WHEEL trucks, which originated on the Pacific Coast, are asserted to be a new and worthy tool that is based on sound engineering principles and meets a real demand for an economical transportation unit.

Although the design has not yet been perfected, the vehicle is making creditable history for itself and the author asserts will be a permanent type. He refers to road-impact tests as evidence of its ability to carry heavy loads with less destructive effect to the roads and its own mechanism than would occur with the four-wheel truck. The industry is urged to conduct an educational campaign and recommend appropriate State legislation that will permit reasonable and just gross weight.

Six-wheel trucks are classified by the kind of axles used, as dead or driving; weight distribution is discussed and mechanical details of present constructions are described at length. Much attention is given to the question of the use of a third differential in the four-wheel-drive bogie to divide or equalize the power to the two sides of the truck, and opposed views of engineers are quoted in an appendix.

Tire wear also is considered at length, with reports on this phase embodied in a second appendix.

The author has followed six-wheel-truck development closely for a number of years and in this paper brings the reader up to date on the subject in an interesting way.

WHEN a device is based on sound engineering principles and fills a real demand, which is reflected in the final analysis by the improved economics of the situation, men should accept it as a new and worthy tool. The six-wheel truck is such a device, and, while many of the principles involved were not so well understood at its inception as they are now, it has been refined to such an extent that its possibilities have gone far beyond the expectations of the pioneers. Much has been written on this subject, and in this paper I shall endeavor to portray the present status of the six-wheeler.

I define a six-wheel truck as a wheeled vehicle incorporating a single, rigid frame-structure for power-plant, driving mechanism, cab and body, carried on three axles, at least one of which drives. This definition draws an immediate distinction with the articulated six-wheeler or commonly-termed "semi-trailer." In view of individual wheel suspension, which may come into the truck field, it may become necessary later to state that the truck is "carried by six wheel-spindles in normally paired transverse alignment." It is possible to build a vehicle with two very narrow-track rear axles in axial alignment, each swiveling at the center or connected thereat with a transverse equalizing bar. However, such a special construction seems worthily outside of the definition, for the present at least.

The number of wheels is purposely omitted to avoid confusion between single or double wheels, or single or dual tires, as a six-wheeler may have anywhere from 6 to 12 wheels or tires. The number of wheels could be limited to the number of hubs or to spindle ends.

Present Status of Six-Wheel Movement

The famous comparative road-impact tests of the Office of Public Roads of the Department of Agriculture are too well known to bear repeating here. While the original six-wheel movement was aimed at reducing the roadbed stress imposed by heavy vehicles, it quickly spread to even light vehicles that could traverse poor roads without tearing them up to the extent as would a four-wheeler with the same load. While these lessened road stresses are vital and should interest our legislators, the truck industry was interested in the reactions of lessened road stress; namely, the decreased shock loads on the truck structure and mechanism. Abnormal constructions and monstrosities were avoided by this sound engineering conception, and destructive road stresses were avoided by the elimination of excessive axle loads, which is the consideration of interest to the highway engineer. I believe that this is one of the finest examples of modern engineering progress.

The soundness of the idea is substantiated by the continually increasing popularity of the six-wheeler, which can be seen on almost every traffic lane in this Country. The truck user is ever on the alert to decrease his cost per ton-mile and is utilizing this type of truck to that end. The enthusiasm thus expressed must not be mistaken as pointing out the six-wheeler as the solution of every transportation problem. The movement has received unjust condemnation in many cases where unsuitable units or unwarranted capacities were recommended. The six-wheel truck is no exception to the rule that each case is an individual problem and should be carefully studied before recommendations are made. The four-wheel truck will always be with us, and the particular conditions of each situation will dictate which type of truck to use. The field is broad enough for all types, and sound, honest recommendations are more vital today than ever before to the maintenance of confidence in our great industry.

There are two fields of service that dictate the general construction of the vehicle and its units. Most trucks are being run over improved roads and their requirements are very different from those of vehicles that must traverse poor terrain such as is found in excavation and construction work, quarries, sand pits, oil fields and mine stripping, and of trucks for transporting bulky loads, often on private property where State or local laws do not apply. For use in still rougher terrain and where commercial considerations are entirely absent, are the military cross-country six-wheelers, which have been developed to a point where only crawler-type equipment would have been considered feasible a few years ago.

Having passed the experimental and commercial production stages, the motor-vehicle industry is now in the throes of unjust burdens fostered by selfish interests.

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TABLE 1—SIX-WHEEL-TRUCK CLASSIFICATION ACCORDING TO AXLE TYPES

Group	Front Axle	Center Axle	Rear Axle
I	Dead	Driving	Dead
II	Dead	Driving	Driving
III	Driving	Driving	Dead
IV	Driving	Driving	Driving
V	Dead	Dead	Driving
VI	Driving	Dead	Dead

We have heard numerous warnings from spokesmen of the industry and they must not go unheeded if we are to exist and continue without retardation in contributing our share to the march of civilization. Apart from the discriminatory legislation with which we all are familiar, I wish to emphasize the injustice done to the six-wheeler in many States where no greater gross weight is allowed than with a four-wheeler. Many progressive States recognize the six-wheeler's virtues and permit increased capacity. The *Commercial Car Journal* for July, 1932, shows that maximum gross loads allowed vary between no restriction, 48,000 lb. and 16,000 lb. Fifteen States give no preference to the six-wheeler over the four-wheeler, and where preference does exist the allowance varies in different States.

In many places today the life of the six-wheeler is threatened. Being based upon indisputable engineering principles and possessing economical advantages as regards the vehicle and its load, as well as the roads upon which it runs, it seems inconceivable that legislators can be blind to these facts. In the Dark Ages, progressives were burned at the stake; in certain centers today, we are spared this bodily incineration but our brain-children are not. The responsibility is on the industry to start an educational campaign on this subject, and I recommend that the Motorcoach and Motor-Truck Activity Committee of the Society prepare a small brochure on the subject of the six-wheel truck, explaining its principles so that the layman can readily understand it and recognize the significance of the results of the Department of Agriculture tests. It should also incorporate recommended legislation, as exemplified by those States that show no favor to the automobile industry but permit what is reasonable and just. This brochure should then be placed in the hands of legislators and motor-vehicle bureaus.

Classification According to Axle Types

The simplest classification of six-wheel trucks is according to the kind of axles used, be they dead or driving, among the necessary three. Table 1 indicates the variety, and I believe the sequence follows the chronological order of commercial development. I am not familiar with any vehicles built under Group VI but they are feasible for special low body-heights where slight grades are encountered and a front drive would be permissible.

To date Group I, with the center driving axle, is enjoying the greatest popularity, comprising both in-built and "attached" rear axles. The simplicity of construction, reduced cost and weight, and lower maintenance make the Group I type desirable where special tractive ability is not essential. Group II, with two rear driving axles, was developed for high tractive ability at the expense of cost, complication and weight and with many paradoxical situations of which Group I is free. Group III, with live front and center axles, as well as Group IV, with all axles driving, were evolutions of the four-wheel-drive truck, combining its virtues with the rear-bogie developments. Group V, with the driving axle at the rear, has been sponsored by the Federal Motor Truck Co. and possesses certain advantages.

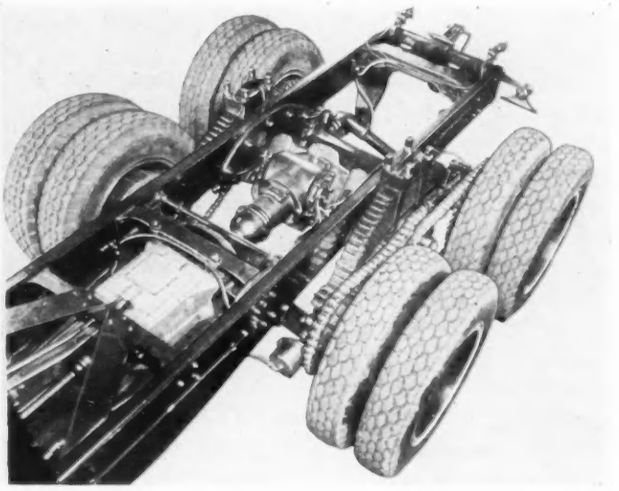


FIG. 1—STERLING DUAL CHAIN-DRIVE

Weight Distribution on Axles

Where the combination of a driving and a dead axle is used in a bogie unit, greater weight usually is borne on the driving axle to increase its tractive ability. With two driving axles, the rear-end weight usually is divided equally between them. The distributions that actually occur will be found in Table 2, and the approximate weight distribution over the three axles, in pounds and percentage, of some current models is given in Table 3.

Axle Constructions and Drive

All forms of mechanical drive known to the prior art are used in the six-wheel truck; namely, chain, worm, double-reduction and bevel-gear. Mack engineers produced a few years ago the double-jackshaft construction[2], which is continued in the same form except that the power divider is placed on the forward side of the front jackshaft and is of a distinctly new and unique type.

The Sterling Motor Truck Co. has been in production for about a year on a dual-chain-drive chassis, utilizing a single jackshaft unit driving all four wheels, as shown in Fig. 1. The axles are entirely free to assume any position required by the unevenness of the road without binding or cramping the ends of the springs. In addition, the springs merely rest on the axles and pivot about the center of the main housing. Each is mounted

[2] See S.A.E. JOURNAL, December, 1929, pp. 591 and 592.

TABLE 2—STATIC-WEIGHT DISTRIBUTION IN BOGIE UNIT

Model	Axle-Type Group (See Table 1)	Weight on Forward Axle, Per Cent	Weight on Rear Axle Per Cent
Dodge	I	55	45
Federal	V	40	60
Hendrickson	II	50	50
Maccar	II	50	50
Mack (Shaft Drive)	II	50	50
Maxi	I	60	40
Sterling	II	50	50
Timken	I	50	50
Timken	II	50	50
Trucktor	I	60	40
Truxmore	I	65	35
(Adjustable to Any One	I	58	42
of Three Positions)	I	50	50
Wisconsin	II	50	50

on a spring seat having a very large bushing provided with ample lubricating chambers. This spring attachment accordingly divides the loads. The center of the spring rotation is also the center of the small driving sprockets. To accommodate the spring deflection, the four longitudinal radius-rods located directly under the springs have ball-and-socket construction at the spring seat and these rods hold the driving wheels in perfect longitudinal alignment with the frame and at the same time permit plenty of flexibility for spring deflection. These rods also have the necessary provision for chain adjustments.

Two cross radius-rods, one for each axle and having a ball and socket at each end, hold the axles crosswise in correct relation to the frame. These two rods are provided with springs that deflect sufficiently to permit the rear wheels to track when making a sharp turn. Therefore, the rear axles are entirely free to follow the roughness of the roadbed, and the wheels can move up and down with relation to the frame without any cramping or binding.

Fig. 2 shows the three-in-one differential used in the Sterling. All of the differentials are located in one housing. The middle differential is the power divider and equalizes or divides the power to the two sides of the truck. Immediately adjacent on each side of the middle power divider is a slightly smaller power divider. The one on the right equalizes the drive between the two wheels on that side by functioning on the tubular and solid jackshafts. The differential on the left acts similarly on the two jackshafts on the left side. Since all of the small sprockets are on separate shafts and correct differentiation is provided, the four chains tighten up when power is applied before the truck moves.

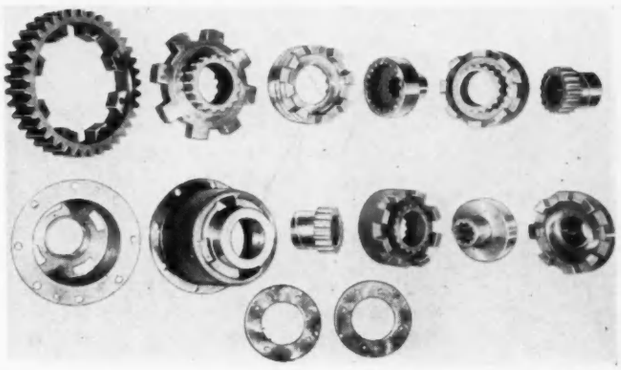


FIG. 2—PARTS COMPRISING STERLING DIFFERENTIAL

The Krohn differential was adopted by the Sterling company and the entire unit is built in its factory. At no point is the drive transmitted through bolts. The bolts that attach the large ring-gear to the differential housing merely hold the two halves together, the drive being transmitted directly through the gear to the very

TABLE 3—STATIC-WEIGHT DISTRIBUTION ON THREE AXLES

MODEL	Axle-Type Group (See Table 1)	Chassis and Cab Weight, Lb.				Chassis and Cab Weight, Per Cent			Gross Weight, Lb.				Gross Weight, Per Cent			Payload and Body Weight on Rear Unit, Per Cent
		Front	Center	Rear	Total	Front	Center	Rear	Front	Center	Rear	Total	Front	Center	Rear	
Autocar	II	6,700	5,300	5,300	17,300	39	30½	30½	9,000	15,500	15,500	40,000	22	39	39	90
Diamond T 801	II	4,235	2,617½	2,617½	9,470	45	27½	27½	5,165	7,917½	7,917½	21,000	24	38	38	92
1201	II	4,650	3,575	3,575	11,800	39	30½	30½	5,950	11,025	11,025	28,000	21	39½	39½	92
1601	II	5,020	4,240	4,240	13,500	37	31½	31½	6,820	14,590	14,590	36,000	19	40½	40½	92
2500	II	5,315	4,407½	4,407½	14,130	38	31	31	7,385	16,307½	16,307½	40,000	18	41	41	92
Dodge																
153-in. WB.	I	1,620	1,285	1,145	4,050	40	30	30	1,800	5,650	4,550	12,000	15	47	38	80–100
158-in. WB.	I	1,870	1,460	1,270	4,600	40	30	30	2,275	6,900	5,400	14,575	16	47	37	80–100
General Motors																
T-95	II	6,150	4,397½	4,397½	14,945	42	29	29	7,400	16,300	16,300	40,000	18	41	41	95
T-95	II	6,150	4,397½	4,397½	14,945	42	29	29	8,655	15,672½	15,672½	40,000	22	39	39	90
Hendrickson																
22-D	II	3,980	2,060	2,060	8,100	49	25½	25½	4,400	8,800	8,800	22,000	20	40	40	80
36-D	II	4,620	3,325	3,325	11,270	41	29½	29½	7,000	14,500	14,500	36,000	20	40	40	80
38-D	II	5,150	4,315	4,315	13,780	37	31½	31½	7,000	15,500	15,500	38,000	18	41	41	80
44-D	II	5,500	4,350	4,350	14,200	39	30½	30½	8,000	16,000	16,000	40,000	20	40	40	80
Kenworth																
346-C	II	6,100	4,410	4,410	14,920	41	29½	29½	8,000	13,000	13,000	34,000	24	38	38	88–90
Maccar																
SW-86	II	5,250	4,075	4,075	13,400	39	30½	30½	8,025	15,337½	15,337½	38,725	21	39½	39½	86–90
TD-500	II	6,775	4,725	4,725	16,225	42	29	29	9,775	17,112½	17,112½	44,000	22	39	39	86–90
Mack																
AC (Chain)	II	6,350	4,425	4,425	15,200	42	29	29	8,900	20,800	20,800	50,500	18	41	41	91
AK	I	5,450	4,650	4,650	14,750	37	31½	31½	8,900	20,800	20,800	50,500	18	41	41	91
AK	II	5,475	5,210	5,210	15,895	34	33	33	8,400	20,800	20,800	50,000	17	41½	41½	91
AK (Light)	II	5,950	4,775	4,775	15,500	38	31	31	8,350	16,700	16,700	41,750	20	40	40	91
AP (Chain)	II	6,500	4,500	4,500	15,500	42	29	29	9,350	20,800	20,800	50,950	18	41	41	91
AP	II	6,200	5,425	5,425	17,050	36	32	32	8,800	20,800	20,800	50,400	18	41	41	91
BX	I	5,520	3,740	3,060	12,320	45	30	25	8,000	15,500	12,700	36,200	22	43	35	91
BX	II	5,550	3,725	3,725	13,000	42	29	29	7,600	14,050	14,050	35,700	21	39½	39½	91
BQ	I	6,000	5,150	4,350	15,500	39	33	28	8,300	18,150	14,850	41,300	20	44	36	91
BQ	II	6,000	4,800	4,800	15,600	38	31	31	8,300	16,450	16,450	41,200	20	40	40	91
Sterling																
FCS-210	II	5,100	5,000	5,000	15,100	34	33	33	9,200	23,850	23,850	56,900	16	42	42	90–95
White																
630 SW	II	4,600	2,950	2,950	10,500	44	28	28	5,250	9,375	9,375	24,000	22	39	39	90–95
642 SW	II	5,350	3,900	3,900	13,150	40	30	30	6,800	13,600	13,600	34,000	20	40	40	90–95
643 SW	II	5,750	4,575	4,575	14,900	38	31	31	7,450	16,275	16,275	40,000	18	41	41	90–95

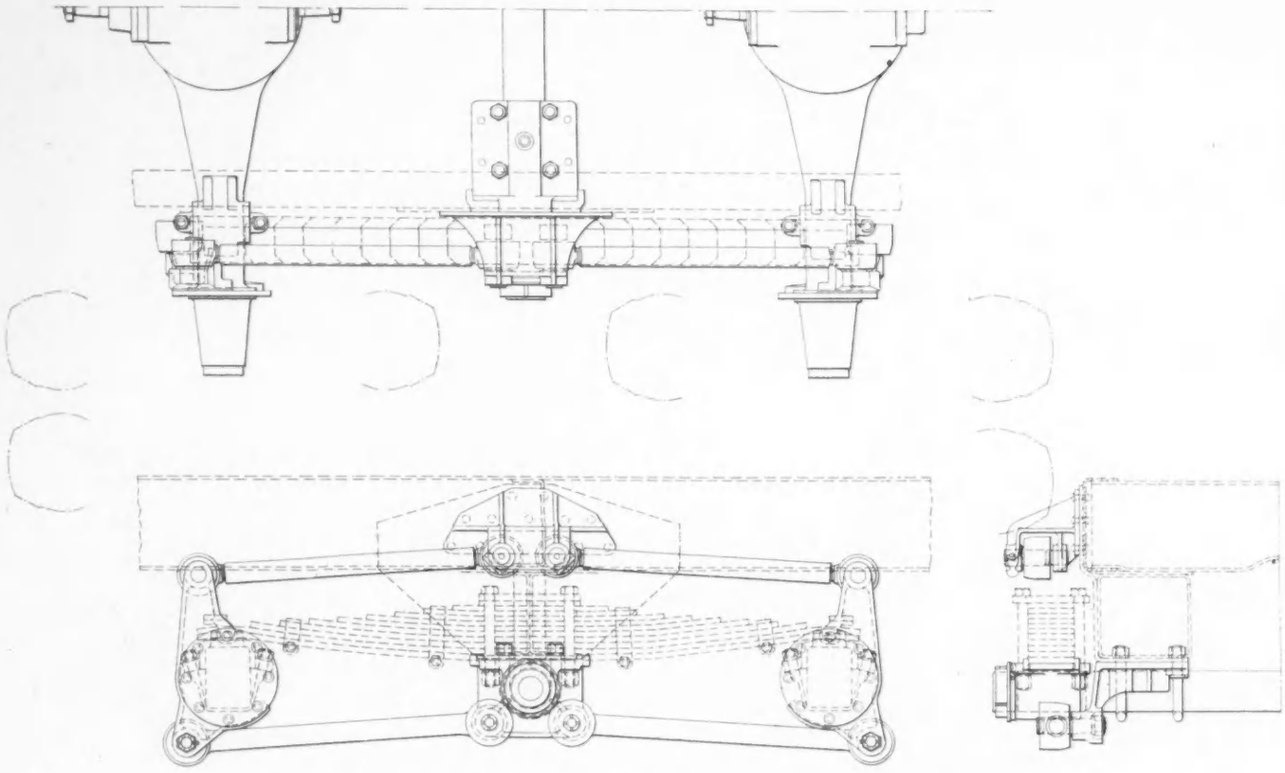


FIG. 3—TIMKEN-DETROIT DOUBLE WORM-UNIT, WITH EIGHT TORQUE-RODS

substantial driving lugs machined on the inside of the gear. The driving chains are special in that they are of a swiveling type and will permit an angle of 4 deg. at each link. This allows the axles to take an angular position with relation to the middle sprockets without causing any binding action in the links.

The Timken walking-beam construction, which characterized the earlier designs and is still in production in improved form for the heavier models, has been supplanted in the new lighter models by the construction shown in Fig. 3, in which the torque is taken by light tie-rods, the working surfaces of which are of the latest type rubber bushings requiring no lubrication. The tie-rods anchor to the spring frame-brackets, which are reinforced by a cross-tube. While a through worm drive is illustrated, Wisconsin double-reduction carriers are interchangeable with the worm units. In the lighter models, the upper four tie-rods are replaced by two centrally located rods, as shown in Fig. 4 on an Indiana 95-SW-75 truck. Timken builds similar units in which a trailer axle of tubular form can be substituted for the rear driving axle. Bevel-gear axles are also available for the forward axle. The entire bogie unit can be fitted with two trailer axles under four-wheel semi-trailers.

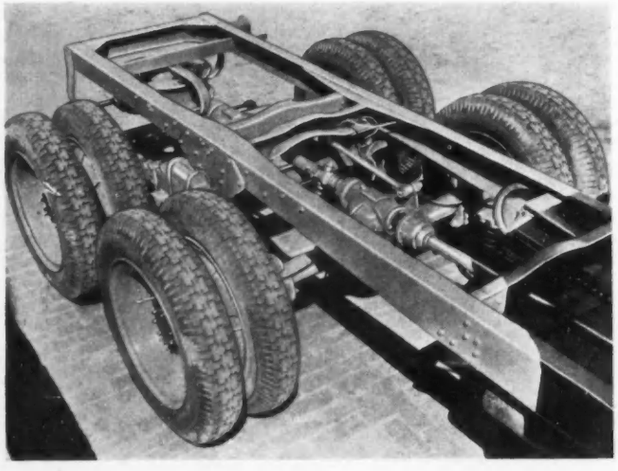


FIG. 4—INDIANA TRUCK EMBODYING LIGHT TIMKEN-DETROIT BOGIE UNIT

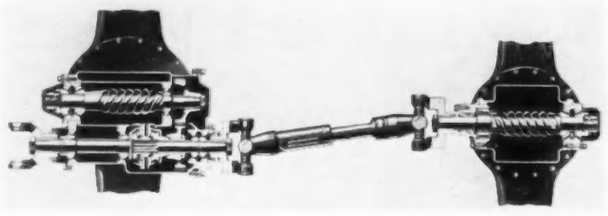


FIG. 5—TIMKEN WORM DRIVE WITH INTERAXLE DIFFERENTIAL

The Timken drive with interaxle differential is shown in Fig. 5 on worm-drive axles. Here again Wisconsin carriers are interchangeable, the forward carrier having a through bevel-pinion shaft.

In the heavy-duty units, Timken has made a number of refinements in design. The lubrication of the main-beam journal is by an oil-wick feed from a reservoir in the cross-tube containing engine oil, as shown in Fig. 6. The spherical portions on the axle housing have been changed to obtain greater area and to have the bronze at all times covered by the steel. An oil-wick feed has

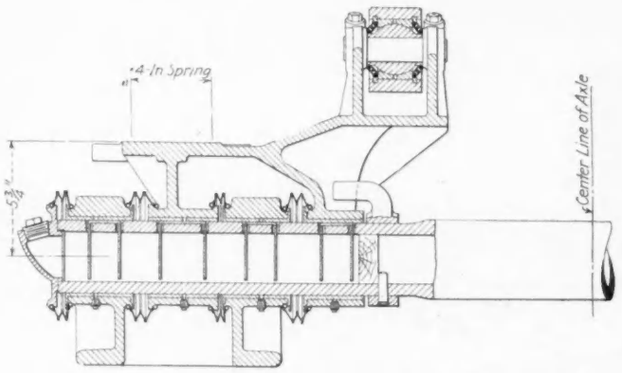


FIG. 6—IMPROVED TIMKEN MAIN-BEAM JOURNAL AND TORQUE-ROD END

been introduced also at this point, using the space between the housing and the sleeve as a reservoir. The torque-rod construction has been changed to a new type similar to that previously used as a radius-rod connection. The design lends itself to precision machining of the bearing surfaces and permits of adequate lubrication.

Hendrickson, a pioneer in the six-wheel field and famous for his "parallelogram" construction, maintains the walking-beam below the center of the axles. Fig. 7 illustrates in the upper view one of his latest bogie units incorporating double-reduction drive and a Krohn interaxle differential contained within a unit ahead of the forward carrier. The pinion shafts are symmetrically offset and a relatively long interaxle propeller-shaft is made possible.

A transfer case, interchangeable axle housings and identical carriers are used in the Hendrickson Model 22-D, as indicated in the lower view of Fig. 7. The case contains a third differential, which is practically identical with the standard types. In the heavier models a manual lock is used, operable from the driver's seat for use in snow, ice or mud. The manufacturer claims that it is less expensive and more positive than an automatically locking differential.

The axle construction is reminiscent of the Relay two-engined truck[3] of Fig. 8 which was experimented with a few years ago. In both cases, the rear-axle propeller-shaft unit has a bearing box supported on the forward axle housing. I am inclined to believe there is a real future for multiple engines in the larger vehicles.

The International Motor Co. has introduced a new line of shaft-driven Mack models characterized by a straight-line through drive. The power divider is mounted on the forward axle housing, as shown in Figs. 9 and 10. The latter illustration is a section through the forward axle carrier. The rear carrier is identical except that the regulation pinion is used in place of the special mechanism shown. The double-reduction axle, with the spur-pinion shaft to one side of the center, permits the use of bevel gears, whereas, in the chain-drive model, the hypoid gearing is necessarily maintained. Torque-arms extend from each housing to a rubber-cushioned support on an upper cross-tube. Individual radius-rods extend from each axle end, adjacent to and outside of the centrally pivoted load-carrying springs, to ball sockets closely nestled to the pivots. The latter are provided with a large bearing surface on each side of the spring, which is clamped into the pivot casting or journal so that the axis is approximately central in the height of the spring. The spring ends are provided with semi-spherical balls that seat below the axle in a swinging shackle-unit swiveling on the axle.

The Federal company is producing the somewhat unconventional four-wheel-drive bogie unit shown in the

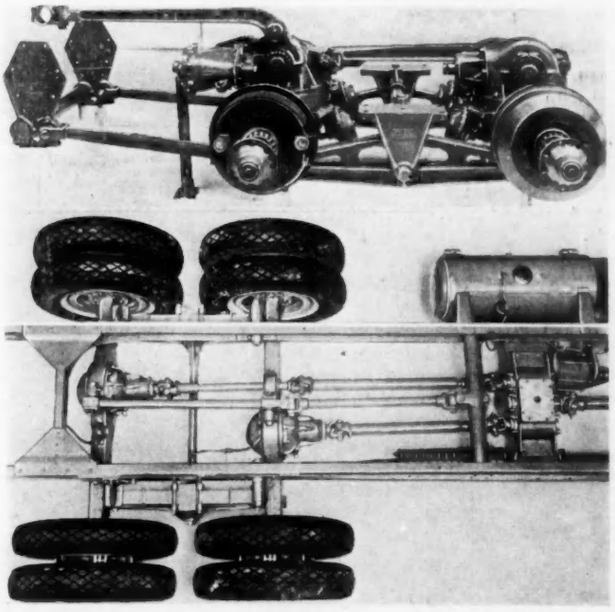


FIG. 7—(TOP) HENDRICKSON BOGIE UNIT AND (BOTTOM) HENDRICKSON DOUBLE DRIVE, WITH TRANSFER CASE

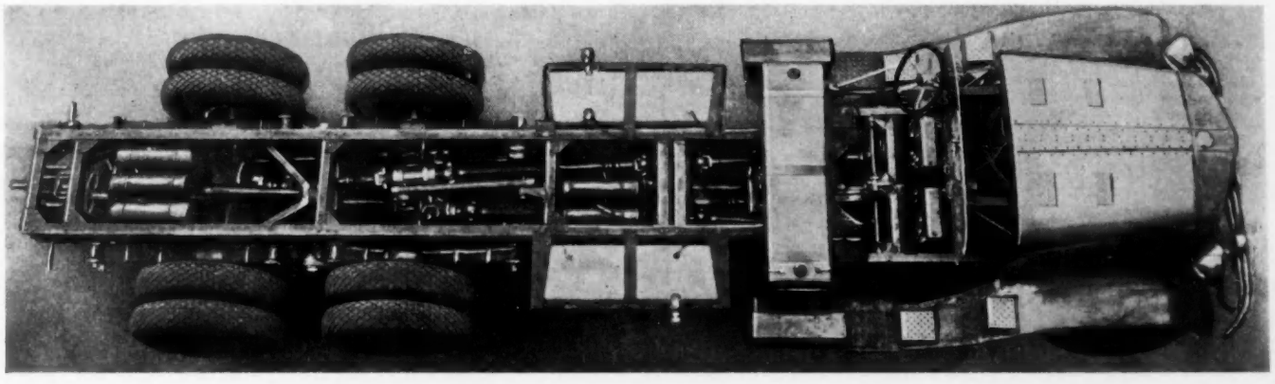


FIG. 8—RELAY EXPERIMENTAL TWO-ENGINED FOUR-WHEEL-DRIVE TRUCK

[3] See *Commercial Car Journal*, January, 1931, p. 38.

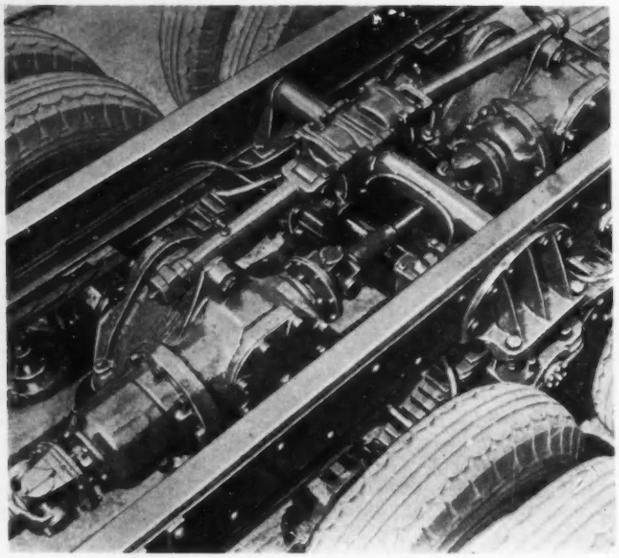


FIG. 9—MACK SHAFT-DRIVEN BOGIE UNIT

upper view of Fig. 11. A rearwardly disposed bevel pinion meshes with the forward-axle ring-gear and drives the rear-axle ring-gear on the opposite side to compensate for the reversed rotation of the intermediate propeller-shaft. The construction is simple and many standard parts can be utilized. Paired, parallel, reversed semi-elliptic springs, characteristic of Federal six-wheelers, are used for loading and torque transmission.

The Federal single-axle-drive bogie is shown in the lower view of Fig. 11, with the trailer axle forward to take advantage of the added loading on the rear axle due to torque reaction. Also, the springs are eccentrated to the rear to increase the static loading of the rear axle. The trailer axle is tubular, of forged nickel steel, with a cast drop-center to clear the propeller-shaft. In the four-wheel-drive construction the springs are eccentrated forward to equalize the variation in traction due to load shifting under torque load.

In the Dodge construction, shown in Fig. 12, the conventional driving axle is at the left and the tubular

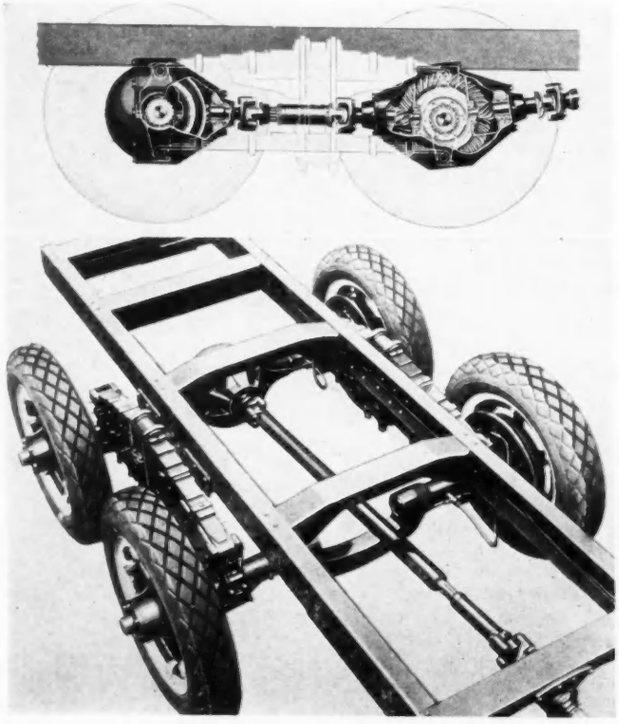
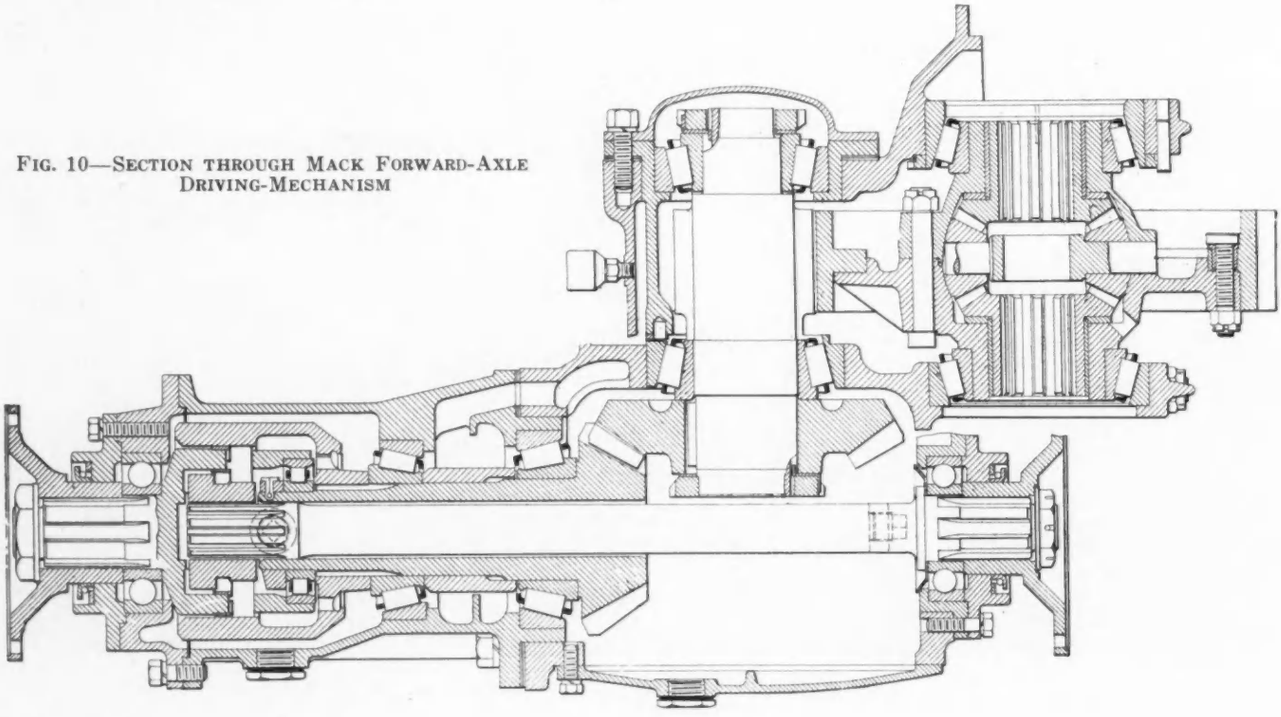


FIG. 11—(TOP) FEDERAL FOUR-WHEEL-DRIVE UNIT AND (BOTTOM) SINGLE-DRIVE UNIT

FIG. 10—SECTION THROUGH MACK FORWARD-AXLE DRIVING-MECHANISM



trailer axle at the right, which is the rear of the truck. The load-carrying spring is above, and pivots on, the cross-tube. It is eccentrated to throw 55 per cent of the static load on the live axle. An end casting on each axle takes the spring ends above center, while ball-jointed torque-rods below extend to a deep vertical gusseted frame-member. The trunnion tube, being hollow, is used as a lubricant reservoir and is filled through the plug in the spring-seat retainer.

Attachments for Converting Four-Wheel Trucks

The six-wheel movement originated on the Pacific Coast, which has fostered many notable transportation developments. It appeared in the form of an attachment, a number of which are still on the market. The attachment is usually in the form of a trailer axle behind the driving axle of a four-wheel truck, with connection and suspension means to form either a bogie unit or an equivalent compensating mechanism to take care of road inequalities. Naturally, the early attempts were more or less crude and experience and use gradually indicated certain violations of physical principles. However, the mere fact that a device is an attachment does not condemn it, as some are prone to assert. For its particular field, if properly designed and engineered

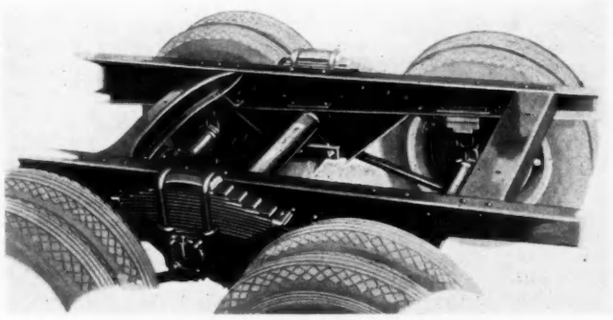


FIG. 12—DODGE BOGIE CONSTRUCTION

into a truck, satisfactory performance can be obtained. In fact, some truck manufacturers have used and are still using such attachments.

The Utility unit, using a walking-beam at each side between the trailer axle and the shackle at the forward end thereof connecting it with the truck spring, retains the same general construction as formerly. The Truxmore unit utilizes a similar walking-beam, as shown in Fig. 13. The most interesting feature is the local-distribution adjuster, whereby the position of the shackle can be varied by screw adjustment so as to give one of three locations resulting in a 50-50 ratio (position shown), 58-42 or 65-35, the first figure being the ratio on the driving axle.

The Trucktor unit in Fig. 14 utilizes a short equalizing beam between the downwardly extending shackles of the regular spring and the trailer-axle spring, being pivoted on the cross-tube member. The location of the pivot hole in the beam is varied at the time of installation to suit the particular requirements of the vehicle. A V-shaped yoke acts as a towing member and brake-torque unit. It is a U-section steel casting and is anchored to the tubular cross-member by a ball-and-socket joint. The spring seats are bushed on the axle, restricting the springs to the sole function of load-carrying members. By using a slipper type of end at the back of each trailer-axle spring, a certain amount of tracking motion is possible. As a rule from 58 to 60 per cent of the static load is placed on the driving axle, although a 50-50 ratio sometimes is used to come within

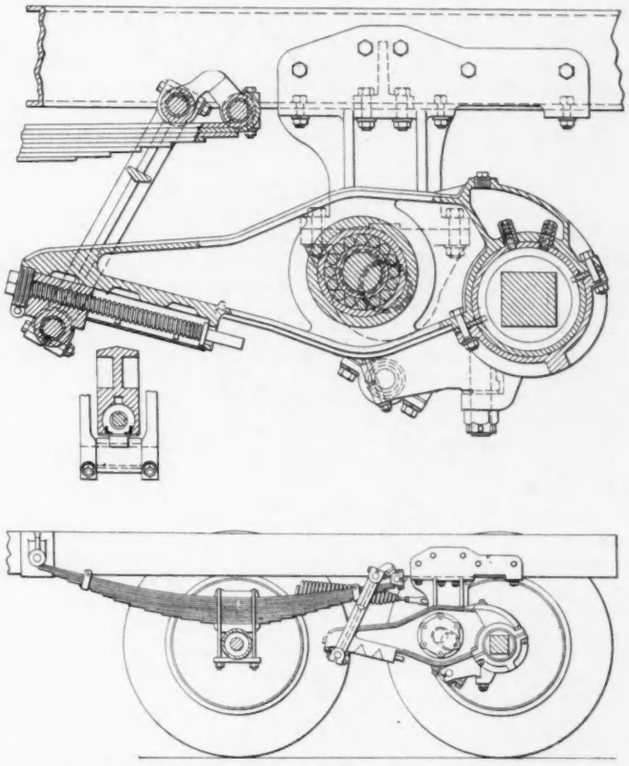


FIG. 13—TRUXMORE ATTACHMENT FOR CONVERTING FOUR-WHEEL TRUCK TO SIX-WHEEL UNIT

the rated tire capacity or not to overload the forward axle.

The Maxi unit incorporates a pivoted, inverted load-carrying spring above the axles, being attached thereto by universally swiveling shackles of the cross-pin type. Torque-rods below the axles are secured to a frame bracket and form a parallelogram with the spring.

The Warford unit for Fords is interesting in that a second Ford axle is placed to the rear of the regular drive axle. A special housing is secured to the front

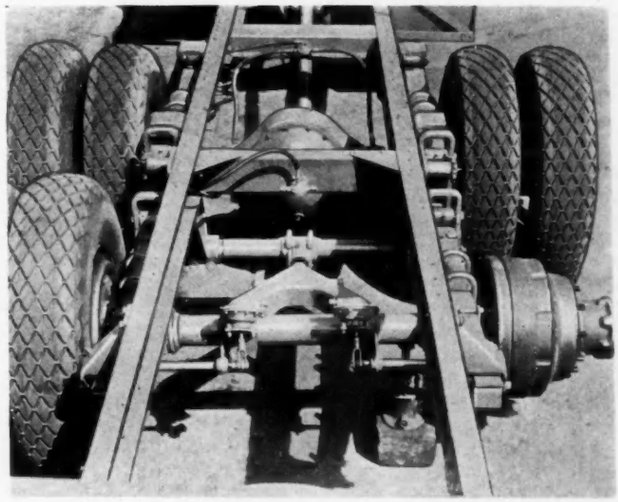


FIG. 14—TRUCKTOR SIX-WHEEL-TRUCK-ATTACHMENT CONSTRUCTION

side of each axle in place of the standard pinion-shaft case and contains a spur-gear drive from the forward-axle pinion-shaft to the rear one through the intermediary of a short propeller-shaft. A gear clutch is provided on the forward gearing to connect or disconnect the drive to the rear at will. A control rod runs forward to the cab.

Wide Variety in Suspension

The method of transferring the weight from the frame to the bogie axles shows the greatest possible variation, ranging from the total elimination of springs to their use in the fullest extent, as exemplified in the parallelogram construction. As the majority of these arrangements were covered in a previous paper[4], I shall confine myself herein to subsequent and improved developments. I have already mentioned the distribution of weight between the bogie axles, and this is more completely covered in Table 2.

Attachments of the original type utilized the regular rear springs and supported the walking-beam pivot by a frame bracket. The Utility and Truxmore units exemplify this type of construction.

As regards frame attachment and relative position, the conventional rear spring is used in the Timken heavy-duty and the Hendrickson bogie units, where the spring seat is swiveled to the center of the walking-beam. The semi-elliptic rear spring in an inverted position and swiveled to the frame is very popular. It is utilized as a pure load-carrying member by Sterling, Timken light-duty units and Mack. In the Dodge and the Maxi unit the spring is combined with the torque-rods to form a torque-resisting parallelogram. Federal utilizes parallel springs to transmit the load and torque reactions.

The entire elimination of springs was sponsored by the Maccar company on one of its models and subsequently by LaFrance-Republic. A walking-beam connects the two driving axles and is pivoted at its center directly to the frame, without any springs. The rocking action of the walking-beam halves for the frame the height of a road obstruction. Further absorption of irregularities comes from the cushioning qualities of the low-pressure tires used. A number of Maccar trucks have been used in severe long-distance work for about two years and have evidently proved the principle to be entirely satisfactory. J. W. Shields, of the Firestone Tire & Rubber Co., has checked these trucks very closely for tire performance and, as far as he can determine, the fact that the springs are omitted does not in any way react detrimentally to the service of the tires.

Torque Reaction

From the foregoing descriptions and figures, the means of absorbing the driving and brake-torque reactions of the various constructions is self-evident. When springs are not depended upon for this purpose, in whole or in part, torque-rods are used in various combinations. In the Mack they are used in conjunction with radius-rods. Timken's pairing of them, above and below the axle, gives a similar parallelogram but with the lower side below the axle centers instead of passing through them. The same company's walking-beam units simulate the Mack parallelogram except that the lower side is rigid. Torque reaction, like the poor, is always with us and changes the static loading on the bogie axles to a varying degree with different designs.

In some layouts a tendency exists to transfer weight from the forward to the rear bogie axle. This change in weight distribution is not of much importance to vehicles operating over improved roads but is vital to those traversing rough terrain, doing cross-country work or operating on soft ground. With some constructions a shifting of weight occurs as soon as a driving torque is impressed on the wheels.

Torque reaction in the complete vehicle—the transfer of some of the front-axle weight to the rear when going forward—is existent in a six-wheeler as well as in a four. The rear-bogie construction, however, decides whether this weight will be divided equally between the two driving axles or be modified by the torque reactions within the bogie unit itself, making an unequal distribution of weight with a decreasing load always occurring on the forward axle.

With an interaxle differential, the driving torque is divided equally between the two driving axles. Without such, a variation in the torque distribution will occur if the front driving axle is about to lift off the ground or if the wheels of both driving axles are about to slip. So far it has been assumed that no resistance has been offered to the motion of the driving wheels themselves and that they have been revolving at uniform velocity. Different conditions are met with, however, when the driving torque is overcoming rolling resistance to the motion of the driving wheels or in accelerating them.

A thorough analysis of the effects of torque reaction on weight distribution is given by W. Steeds in an article on Rigid Six-Wheeled Vehicles[5] and is recommended highly to everyone interested in this subject. A summary of the analytical results of that article is reproduced in Table 4. The assumptions, based on an actual vehicle, with dead front axle and a pivotal point in the center of the bogie wheelbase, are as follows:

Weight of Vehicle and Load, lb.	12,000
Wheelbase of Truck, ft.	12
Center of Gravity Ahead of Bogie Trunnion, ft.	8
Wheelbase of Bogie Unit, in.	42
Effective Wheel Radius, in.	15.6
Torque on Two Axles (engine torque, 125 lb-ft. x over-all gear ratio of 40:1), lb-ft.	5,000

The great variations shown in Table 4 are no doubt in excess of those that have been commonly supposed. In the examples given, all of which are based on an equal static-load distribution on the two axles, Type I consists of a construction in which the forward axle is anchored to the frame by a torque-tube with a ball-and-socket joint and the driveshaft is above the axle-shaft axis. The rear axle is similar and anchors, by its torque-tube and ball, in a socket at the back of the forward axle. The driveshaft unit is within the torque-tubes and joints. In Type No. II the springs are pinned at their ends to the axle casing and rigidly fixed at their centers to a trunnion block pivoted to a frame cross-member. This type is similar to the Federal unit at the top in Fig. 11, in which it will be noted that the pivot is placed ahead of center, amounting to a change in the lever arm and thus compensating for the increased loading on the rear wheels.

In Type III the driving wheels are mounted at the ends of levers, which are pivoted at their centers on an axle casing similar to that of an ordinary rear axle, the levers taking the place of the usual wheels. The levers, as in the Scammell construction, are casings enclosing a train of gears whereby the axle shafts drive the road wheels.

Type IV is similar to Type II so far as the spring ends are concerned, but the springs are independently pivoted at their centers to frame brackets, the pivot centers and spring eyes being horizontally in line. The parallel springs in Type V are rigidly secured at their centers to a trunnion block pivoted on a frame cross-member. The axle casings are left free to turn inside the members connecting the springs at their outer ends.

[4] See S.A.E. JOURNAL, December, 1929, p. 589.
[5] See *The Automobile Engineer* May, 1931, p. 172.

Torque-rods are connected at one end to the axle casing and at the other end to the frame. The trunnion axis is assumed to be at the same level as the axle shafts. Substituting a walking-beam or radius-rods for the parallel springs, this type simulates the Timken heavy-duty and Mack designs. In both Types IV and V, the torque reactions result in only horizontal forces in the springs for the former and in the torque-rods in the latter and do not affect the equality of the vertical forces acting at the ends of the springs.

Type VI incorporates two similar driving axles, each having a torque-tube connected to the frame by the usual ball-and-socket joint. The Büssing is an example of this method, which also assures equal vertical forces.

Steeds' figures are mathematically correct but are based upon theoretical ideals that do not exist in practice. Steeds assumed that the horizontal torque reactions cannot affect the vertical forces at the ends of the springs or their equivalent parallelogram constructions. This calls for a frame, brackets and all torque-reaction rigging to be absolutely rigid. Who has not seen a frame deflect under a severe driving or braking load? Considerable time might be spent in calculating deflections, but it would not be worth while in view of the innumerable variables involved. The frame bracing by the body will depend upon how the body is attached, or, as in the case of a dump body, very little stiffening is given by the body foundation.

It would be worth while for someone to make a practical experiment in measuring the over-all deflection of the bogie axles from the horizontal in complete trucks where the theoretical ideal shows equal loading on the two axles. Automatic recorders could be used to measure simultaneously the relative positions of the three axles, their relation to a rigid bar correctly mounted on the frame, and the driving torque being impressed on the axles. So long as torque is impressed on the axles, the rear axle will be depressed and the forward one raised, the distance depending upon the over-all flexure under the given impressed torque. In all of the results in Table 4, the rear-axle loadings will actually be greater.

Interaxle Differentials

Considerable argument is presented by the exponents of the interaxle differential as well as by those who do not employ any. Each construction has its field and particular requirements; nevertheless we find much contradictory evidence in the field, where lower maintenance costs are claimed by one over the other. No difficulty should be met in obtaining authentic comparative data on this subject as well as on the unit with a driving and a trailer axle, since interchangeable bogies for the three types are available and accurate field data could be compiled from trucks working under identical conditions. I believe that it would be worth the while of the Transportation and Maintenance Activity Committee to inaugurate such a step so as to define more closely the scope and limitations of each type. Tire wear is usually a good barometer of general maintenance costs, because, if the tires are properly maintained, any mechanical defects will "take it out" on the tires as well as on the general mechanism. This will be discussed under the heading of Tires and Their Performance, and the evidence presented will indicate a wide divergence of opinion.

Variation in the rolling radius of the tires will, in the absence of the interaxle differential, immediately cause one axle to "fight" the other, resulting in mechanical and tire wear. A third differential would compensate for this and eliminate close pressure checking and matching of tire diameters. However, the proponents of the through drive claim that tires can readily be maintained and thus overcome any possible trouble on this score.

Should a forward wheel, for instance, pass over an obstruction while the rear wheel is still traveling on

TABLE 4—TORQUE REACTIONS FROM BOGIE UNIT

(Results of Steeds' Analytical Research)

Wheels Revolving at Uniform Speed. No Resistance to Driving Wheels Themselves

Torque Assumed Equally Divided between the Axles. Value of Torque 5,000 Lb-Ft.

Type No.	R_1 Value	R_1 Alteration, Per Cent	R_2 Value	R_2 Alteration, Per Cent	Remarks
					Minimum Value of μ
I	5,530	+38	3,350	−16	0.58
II	5,860	+46.5	3,000	−25	0.64
III	5,260	+31	3,830	− 4	0.5 n=2
IV, V and VI	4,640	+16	4,640	+16	

Torque Equally Divided. Front Driving Wheels About To Slip. μ=0.6

Type No.	R_1 Value	R_1 Alteration, Per Cent	R_2 Value	R_2 Alteration, Per Cent	Torque, Lb.–Ft.
I	5,570	+39.2	3,330	−16.8	5,180
II	7,350	+84	3,880	− 3	6,050

Torque Unequally Divided. Front Driving Wheels About To Lift. R_2=0

Type No.	R_1 Value	R_1 Alteration, Per Cent	Torque Required, Lb.–Ft.	Minimum Value of μ Necessary
I	10,720	+168	8,850	0.64
II	Conditions not possible			1.35
III	Conditions not possible			2.7

Torque Unequally Divided. Both Pairs of Driving Wheels About To Slip. μ=0.6

Type No.	R_1 Value	R_1 Alteration, Per Cent	R_2 Value	R_2 Alteration, Per Cent	Torque Required, Lb–Ft.	Remarks
I	8,440	+111	620	−84.5	7,000	.....
II	6,690	+ 67	2,560	−36	7,200	.....
III	5,860	+46.5	3,730	− 6.8	7,480	n=2
IV and V	5,000	+25	5,000	+25	7,800	In these cases the torque is equally divided

No Tractive Effort Produced. Whole Torque Used in Overcoming Resistance to or Accelerating Driving Wheels Torque Equally Divided. Value 5,000 Lb-Ft.

Type No.	R_1 Value	R_1 Alteration, Per Cent	R_2 Value	R_2 Alteration, Per Cent
I	4,790	+19.2	3,370	−15.8
II	5,430	+36	2,570	−36
III	4,820	+20.5	3,390	−15
IV, V and VI	4,210	+ 5	4,210	+ 5

n=ratio of drive=revolutions per minute of axle shafts divided by revolutions per minute of road wheels
R_1=rear-axle load
R_2=center-axle load
μ=coefficient of friction (adhesion)

the level, the former must traverse a greater distance which, without an interaxle differential, can be compensated for only by a "breathing" action in the torque-rod hook-up or flexure in the driving-mechanism parts and the tires, which, of course, is allowable if not overstressed. Possible slipping and scuffing of tires and the stressing of parts will naturally depend upon the variation that would occur between two free wheels under such conditions. Where such a variation is slight, an interaxle differential seems superfluous but, in any kind of rough going, it is absolutely essential. Everything hinges on the definition of "rough going." No argument can be presented against the fact that, without an interaxle differential, back-lash is not taken up until creep does it, during which time one axle does all the driving.

The interaxle differential will overcome any of these difficulties but at the expense of weight, complication and initial and maintenance cost. The belief that each has its particular field appears reasonable. However, this is under the assumption that we have an ideal differential, because the shortcomings of the usual type present disadvantages that outweigh the advantages. Slippage of one wheel makes the entire bogie unit inoperative. In a through drive, stalling is impossible if the two wheels on one axle have positive traction. For such reasons, special differentials or substitutes have been developed, such as the Timken High-Traction type and the Krohn compensator.

Assuming that stalling is thwarted, a difficult problem still remains. Should the differential place the full torque on one axle, its driving mechanism must be as strong as that of a single axle which would be substituted for the bogie unit. An equal-power-distributing differential allows each axle to take its half of the load, thus reducing the size of the parts and their weight. How can this paradox be solved?

Many attempts have been made to develop substitutes for the differential by introducing internal resistance or opposing forces to nullify the torque-balancing properties while still retaining the motion characteristics of a true differential. These devices are often wasteful of power, the surfaces become polished and their locking becomes less positive, destructive heat or erratic rotation results and their rapid wear leads to back-lash, which causes destructive impact.

Mack engineers have developed the power-divider shown in Fig. 10 and the upper view of Fig. 15 as an attempt to solve this problem, the latter illustration showing its application to the chain-drive model. The drive from the transmission is delivered to a driving-cage carrying two sets of radially sliding plungers. They make contact on the inside with a double cam, one cam for each row of plungers. The cams are staggered and the use of two rows of plungers assures positive drive, due to the opposing forces. The inside cam is connected to the rear axle or jackshaft. An outside corresponding cam surrounds the round-ended plungers and drives the forward unit. In Fig. 10 the outside cam drives back to the pinion, and in Fig. 15, forward. Current production, however, has been changed to place the power divider forward, as in Fig. 10. The shape of the cams can be varied so that equalization is comparable with that of the standard type of differential. They can also be made for the other extreme of complete locking. The Mack designers have steered a middle course, maintaining sufficient traction on one set of wheels to move the truck, regardless of conditions under the other set, while stopping short of a degree of locking that would entail undue torque loads. Large bearing surfaces characterize the design, and thrust reactions are self-contained because of the concentric relationship of the parts.

There has been considerable argument regarding power loss when the interaxle differential operates. Such a loss would be difficult to calculate, as gearing efficiency is considerably higher today than our antiquated formulas indicate. A rare opportunity exists for someone in the industry to run or sponsor a test to obtain definite figures based on modern tooth forms, hardened gear surfaces and rugged mountings, with substantial alignment. When the differential does not operate, no energy can be dissipated in friction; when it does operate, the relative speeds are low and, if the differential characteristics are such as to allow fairly uniform motion, the loss must be very slight. In view of some complicated gear trains that have tested more than 90 per cent efficiency, I feel that any interaxle power losses due to the differential are negligible, insofar as they would be reflected in truck economy and, further, because the differential is in operation during a relatively small period.

Probably no subject has given rise to a greater conflict of opinion than has the use or omission of a third differential. A symposium of divergent views is given in Appendix 1.

Tires and Their Performance

With single tires on the front wheels of the truck, the wheels of the bogie unit can be equipped with either single or dual tires, or dual tires on the forward axle and singles on the rear. Assuming that all the tires are of the same size and individual tire loadings are the same, we would have the following weigh distributions over the three axles: 33⅓ per cent on each; 20, 40 and 40 per cent and 25, 50 and 25 per cent on first, second and third axles respectively. Tire manufacturers would be delighted if such distributions were always possible, but some compromise is often necessary. Greater mileage will be obtained if the departure from the ideal is on the side of underloading rather than overloading. How the ideal distributions compare with practice will be noted in Table 3.

Driving arrangement also is considered in tire arrangement, in view of traction or road conditions, such as ruts. One manufacturer of four-wheel-drive trucks who adds a trailing or power axle, depending on the type of six-wheeler, prefers single tires on all wheels. However, some customers demand dual tires in the rear, and the wheelbase is rearranged to compensate for the capacities of the different tire arrangements. The builder is also restricted by legislation limiting weights on the axles, and a compromise must sometimes be made on this score.

Group II in Table 1 can be subdivided into units without and with the interaxle differential. We then have the following three types of six-wheeler about which all controversies on live and dead axles and third differentials seem to center:

(1) One driving and one trailing axle
(2) Without interaxle differential
(3) With interaxle differential

If conditions permit, the first type is desirable because of its simplicity and low upkeep cost. It is very popular in certain fields and the compromises seem fewest. Such a truck is representative of several in New York State carrying milk and farm produce to the extent of a 15 to 18-ton payload, with a piston displacement of slightly more than 500 cu. in., chassis weight of 12,000 to 13,000 lb. depending on tire equipment, and making a 450-mile road trip on schedule time, day after day, averaging 20 to 22 m.p.h. over the road. This is a favored size for the first type in this particular service, and the engine has sufficient power to make undue shifting on hills unnecessary, thereby avoiding excessive fuel consumption.

Successful truck operation can be ascertained to a large extent by tire performance. The experiences of operators quoted by the Goodyear Tire & Rubber Co. in Appendix 2 are of interest.

General Design and Requirements

Engine capacities must measure up to the work that is required. The trend is toward larger engines and rightly so. The increased fuel consumption in low-gear work will soon outweigh the possible economy of a smaller engine. As in all types of service, a reserve is essential for long life. An engine of relatively slow speed with high torque output is desirable.

Clutch and transmission must necessarily be capable of taking care of the high engine output. Propeller-shafts must be long to reduce the universal-joint angularities. Much can be done along this line, particularly with the interaxle shaft. To secure the maximum length between the joints, the pins must be brought as closely as possible to the pinion shafts. Some joints of the older type do not meet this requirement, and assembly methods must be changed. The Spicer Mfg. Corp. has produced a new type in which the end yokes are first assembled on the pinion shafts; the joint proper is then installed by inserting the journal crosses in the yokes by tilting, followed by centralizing them with the bearing units and locking them in with end plates.

One of the standard bogie units having double-reduction drive and an interaxle differential imposes a normal driving angle of 10 deg. on the joint, with a maximum interference angle of 42 deg. Such an angle is ruinous to any joint of the plain-bushing type. For this reason spindle or roller bearings are becoming obligatory for this service.

The axles have a greater rotational movement about the axle-shaft axis than in the case of a four-wheeler, and the universal-joints, radius-rods and suspension system must be laid out in accordance with this fact. Shackles, if used, must be long to allow greater oscillation. They and the spring bolts should be more rugged, in view of the more severe duty imposed upon them.

Frames should be reinforced to take care of the additional length at the rear and to provide torsional rigidity in the horizontal and vertical planes for the bogie unit mounting, and especially to resist flexure from torque reaction. The axles must have complete freedom to rise or fall at either side, this being provided by spherical or cross-pin joints or by curved spring-seats where the springs are not pinned. The General Motors rear-spring sliding-block mountings shown in the lower view of Fig. 15 were designed to cope with the greater road irregularities that are expected with a six-wheeler.

The driving mechanism in the axles must measure up to the extreme stress possible, depending upon such conditions as are imposed by one or two driving axles, a third differential and the torque-delivery characteristics of the latter.

The brake situation has improved considerably. Much harm was done by the early six-wheel movement when inadequate or improperly matched brakes were used on the trailer axle. This essential feature has not been entirely overlooked in today's equipment. Equal braking all around at the road contact is desired, and the mechanical laws governing this situation are the same as we found in the study of driving-torque reaction, the only difference being in the opposite direction of the forces. Constructions the same as, or equivalent to, those of Steeds' Types IV, V and VI will show a theoretical equality of braking effort and its reaction by both axles. But here again the flexure of the frame, brackets and torque-reaction rigging nullifies the ideal. We all know from visual inspection that the front wheels "dig in" on braking. That they should not can be proved theoretically, but the ideal rigidity is upset

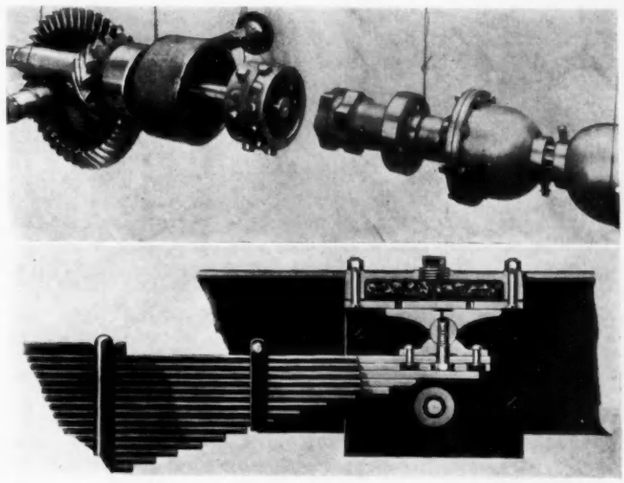


FIG. 15—(TOP) MACK POWER-DIVIDER PARTLY SEPARATED AND (BOTTOM) GENERAL MOTORS TRUCK REAR-SPRING SLIDING-BLOCK MOUNTING

far more by braking stress than by driving torque, because of its greater severity.

If trucks could accelerate as fast as they decelerate, we should see the rear wheels digging in as much. We should be thankful that they cannot, for we should have to fill up the load space with horsepower. Under such braking stress, and because of the downward frame movement, the deflection of the front springs will add their quota to the disturbance of over-all flexure. We have a resilient support under a flexible beam. Since this upsetting flexure is in a different direction for driving than for braking, it would be impossible to offset the bogie unit to suit the one condition, as this would aggravate the trouble for the other. An automatic compensator is conceivable, but to make it simple and dependable will be a problem.

Identical brake equipment on all four wheels is not capable of giving each wheel equal braking capacity. Mack designers seem to be the only ones, so far, who have appreciated this condition and met it by using a large Westinghouse air-brake chamber to actuate the camshafts of the forward-wheel brakes and a smaller chamber at the rear, where less traction is available.

While very little scuffing between corresponding fore and aft rear tires occurs when making a turn, amounting to even less than on adjacent dual tires, its elimination, if possible without involving complication, is desirable, because the steering ability of the vehicle probably would be improved. As will be recalled, the cross radius-rods in the Sterling construction are provided with springs sufficiently flexible to allow the wheels to track when making a turn. The Trucktor attachment shown in Fig. 14 also provides this tracking feature. The connections of the bogie unit to the frame can be made to allow relative lateral displacement of the axles by placing the connections so as to be inclined to the longitudinal axis of the frame instead of parallel[e]. All of these methods shorten the effective wheelbase and consequently the turning radius.

An essential when traveling straight ahead is that the two axles be parallel. When one or two springs are used at the side to tie the axles together, their deflection will increase or shorten the bogie wheelbase. Therefore the truck must be loaded evenly in a transverse direction so that the axles will be parallel. Spring deflection at one side, from a road obstacle for instance, will give a similar effect. One compensating feature of this construction is that, if the distance increases, cen-

trifugal force in going around a turn will give the unit a tracking effect. However, constructional, road, speed and other variables make its utilization practically impossible.

In placing the mechanism of the bogie unit under the truck, space for equipment is sometimes very cramped or entirely omitted. This point should not be overlooked. Other constructions have been engineered to avoid any such conflict. The installation of a direct-acting hoist with vertical cylinders, for example, is impossible without making minor alterations to the bogie. If the cylinders have room, sometimes not enough space is available for the plunger ends to travel in an arc about the rear hinge, so installing auxiliary arms becomes necessary to prevent the swinging of the hoist cylinders in their customary path.

The Six-Wheeler versus the Semi-Trailer

This paper would not be complete without spiking the fallacy, still prevalent in some quarters, that "a vehicle can pull more than it can carry." The difference in first cost of the units and in weight distribution over the three axles is very little when considered on an equal-payload basis. This is easily checked by comparing comparable current equipment. When equal performance is to be obtained, there can be no difference. On an unequal basis, the excessive semi-trailer gross load places an undue burden on the engine. An undersized, overloaded engine furnishes no logical or fundamental basis of comparison but is only its worst enemy, to the detriment of the engine builder and the befuddled operator. Builders of four or six-wheel trucks would not tolerate such a misproportioned vehicle, even though it can be built.

The six-wheeler and the semi-trailer are not competitors; each has its own field of usefulness. The choice of either depends on such items as loading time and facilities, the use of one tractor with several semi-trailers, special-body length, maneuvering ability under certain conditions, legislation favoring one type over the other, and so forth. If the operator will carefully consider his particular requirements and insist upon equal operating performance, either type of vehicle will do the work. However, local conditions will dictate the type.

One point in favor of the six-wheeler over the semi-trailer is its stability. No danger of "jack-knifing" exists, and the truck has greater inherent stability on icy and crowned roads. Considerable work has been done to improve these conditions on the semi-trailer, but a number of problems still remain to be solved to make it as safe for high-speed work as the unit vehicle.

[6] See United States Patent No 1,743,892; Jan. 14, 1930.

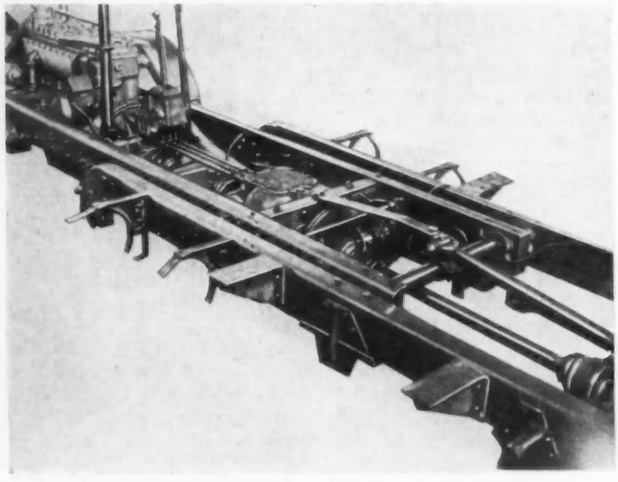


FIG. 16—UNITED STATES ARMY TRUCK CHASSIS, WITH OVERLAPPING DOUBLE FRAME

Six-Wheel-Drive Trucks

Six-wheel-drive trucks are not expected to be produced in any quantity, but a demand for them will arise where extremely bad terrain must be traversed. The present status of this type of vehicle is due entirely to the pioneering work done by the Army at Camp Holabird under the able leadership of Col. E. S. Stayer. The industry owes him a considerable debt of gratitude for his vision and excellent engineering judgment in working out the problems encountered in both the six-wheel and four-wheel-drive vehicles of the six-wheel type. The industry has benefited considerably from his developments and pioneering, one instance being the through-drive double-reduction method.

Some years ago Colonel Stayer predicted that the major portion of military transport would drive on every wheel provided on the chassis. He had decided that a six-wheeler having any dead rear axle decidedly had no place in military activities and that one with a dead front axle was limited mainly to improved roads.

The ability of these Army trucks to perform under the most adverse conditions is remarkable. The 7½-ton six-wheel-drive chassis has transported 10 tons of payload up a 65-per-cent grade, towing a 4-ton gun. The 10-ton chassis has traversed the most hazardous terrain, transporting an 11-ton payload and towing a 13-ton gun. Every multi-wheel-drive job equipped with front-wheel drive has been called upon to readily negotiate a 65-per-cent grade. The Army has conclusively demonstrated that the addition of front-wheel drive to six-wheelers adds a good 50 per cent to the ability of the truck. A 2-ton six-wheeler having selective front-wheel drive was produced. The driving mechanism can be readily engaged at will, at practically any vehicle speed. This vehicle easily negotiates the 65-per-cent test hill as a six-wheel drive. No comparable ability as a four-rear-wheel drive could be obtained. Over bad going, as a quadri-drive, the truck fails to traverse terrain easily covered as a six-wheel-drive vehicle.

Of particular interest is Colonel Stayer's ideas on differentials, as quoted in Appendix 1.

Engine sizes are naturally larger in Army trucks than in similar commercial models, although these are being stepped up to give higher road speeds to long-distance trucks. At Camp Holabird the Quartermaster Corps is endeavoring to provide about 70-lb. rim pull in high gear. A combination transmission and transfer case has been developed that employs three shafts and has no more than two tooth contacts in any forward speed. Low-gear ratios in the 2-ton group range around 45:1. In the heavier models, ratios around 100:1 have been found satisfactory for all-round operation. Deep frame sections of heat-treated chromium-nickel steel are used and follow standard construction except in one instance. The experience may suggest a remedy for similar impact problems. In a combat-tank carrier, a frame sag of 2½ in. developed after several months' test. In this vehicle the tank loads itself over a combination tailgate ramp, attaining a steep angle in the operation, and only an experienced operator can rock it down in place on the truck-body floor without its striking a terrific shock blow. As experienced operators were not always called upon to load this equipment, students sometimes being permitted to practise this feat, foolproofing the design was deemed necessary. Therefore the overlapping double-width frame portrayed in Fig. 16 was developed. The single top torque-rod from the for-

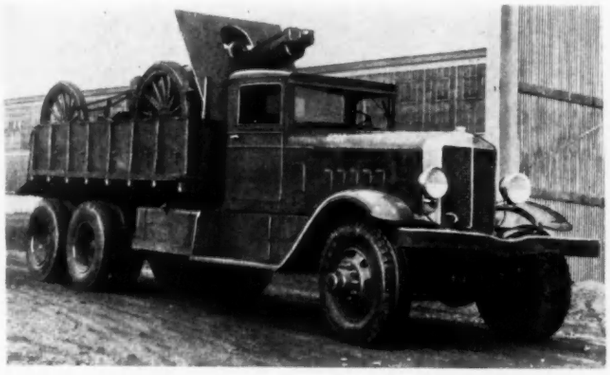


FIG. 17—TYPICAL MILITARY SIX-WHEEL-DRIVE TRUCK

ward axle is universally anchored to the tubular cross-member shown. Radius-rods below the frame at each side parallel the frame members. Double-reduction axles are used in all chassis from 1½-ton capacity up. The front-driving-type axle is a front-mount bus type, while the rear axles are of the top-mount style.

The Army six-wheelers have been set up in groups of the capacities shown in Table 5. A typical six-wheel-drive truck is depicted in Fig. 17.

As a result of the Army's activities, several truck manufacturers are now in production on six-wheel-drive trucks. The Wisconsin Axle Co. has developed a line of front and rear driving axles providing interchangeability of many of the vital parts. Rzeppa joints are used on the front axles. One of the latest Wisconsin designs is shown in Fig. 18.

For work in the foreign oil fields, the Four Wheel Drive Auto Co. has developed a 15-ton-payload model, weighing 20,945 lb. and having an over-all length of 27 ft. 2 in. It is equipped with an 8 x 18-ft. steel body. The regular FWD front axle is used in combination with Wisconsin double-reduction rear axles and bogie rigging. Chassis and cab weigh 18,000 lb., and 12.75 x 20-in. single tires are used all around. The front-axle torque reaction is taken through the front springs.

Conclusion

The six-wheel truck will be a permanent type. In its various forms it can cope with a great variety of operating conditions. The different general types are not competitive but individually fill fields for which they are best adapted. These fields should be clearly recognized so that the six-wheeler will not be condemned because of misapplication.

TABLE 5—ARMY GROSS-WEIGHT RECOMMENDATIONS FOR SIX-WHEEL TRUCKS

Group	I	II	III	IV	V
	Under Cross-Country Operation, Lb.				
Six-Wheel Drive	12,000	20,000	27,000	34,000	40,000
	Under Good-Roads Operation, Lb.				
Four-Rear-Wheel Drive	12,000	20,000	27,000	34,000	40,000
Six-Wheel Drive	13,500	22,000	29,000	36,000	42,000

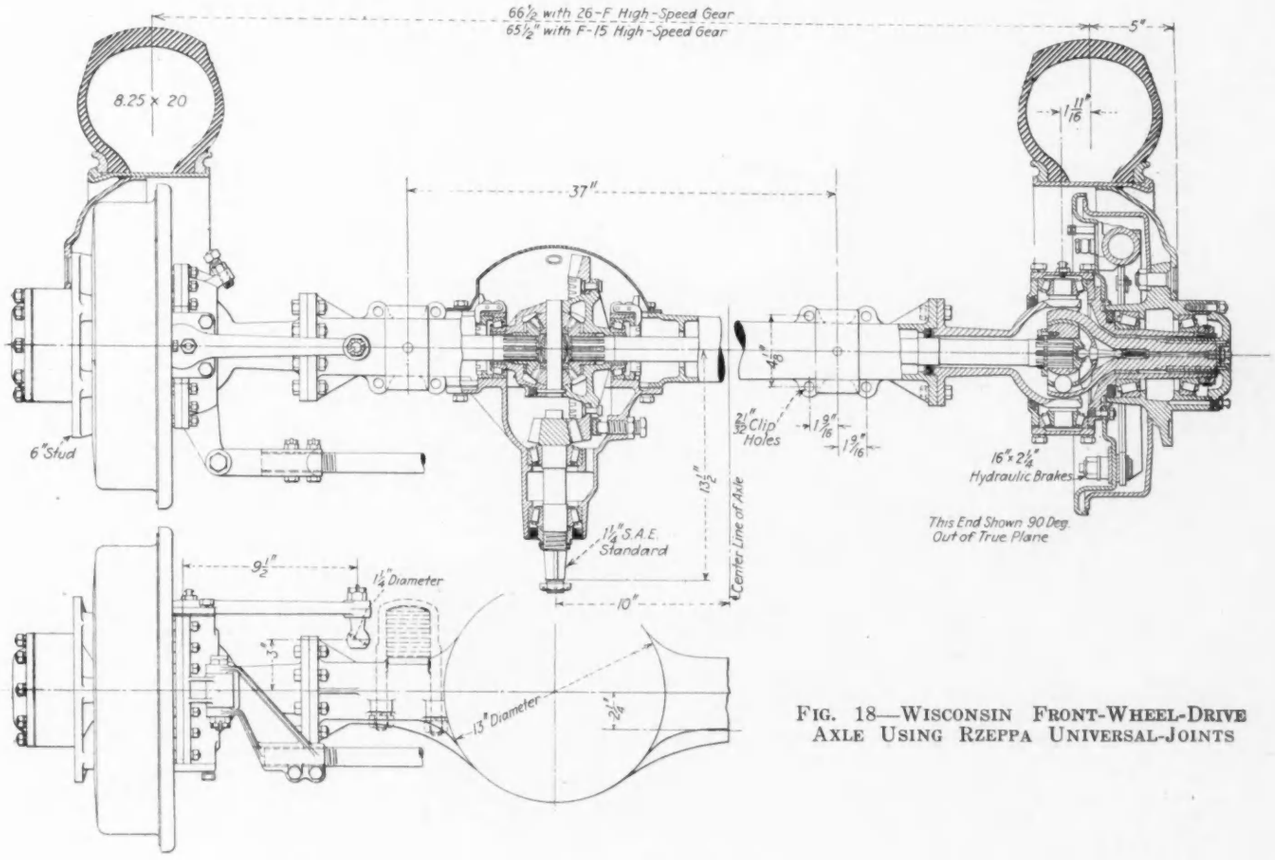


FIG. 18—WISCONSIN FRONT-WHEEL-DRIVE AXLE USING RZEPPA UNIVERSAL-JOINTS

The six-wheeler is not perfected; much remains to be done, as with most other automotive units. However, it has taken its place in the modern transportation field and is making creditable history for itself. My hope is that this paper will serve to stimulate further thought on the subject and that such study will culminate in still better six-wheelers in the future.

I desire to acknowledge my indebtedness to the many engineers who have cooperated so wholeheartedly in supplying me with data and information for this paper.

APPENDIX 1

Engineers' Views on Third Differential

H. D. Church (White Motor Co.):—We have never used a third differential with any of our six-wheel units and have not as yet found any necessity for it; neither have we used any locking or unconventional differentials in any of our six-wheel units with two driving axles. Our field experience with the six-wheel double-driving-axle construction to date has been satisfactory as regards tire wear, and we find that we can carry more torque in a given driving axle in a six-wheel hook-up without any interaxle differential than we could carry in the same axle on a four-wheel installation. This probably is explained by a certain amount of tangential flexibility in the tires, which permits part of the torque applied to the No. 1 axle to be transmitted through to the center of the No. 2 axle.

W. D. Reese (General Motors Truck Co.):—All of our six-wheel vehicles having two driving axles use an interaxle differential. We find this somewhat more satisfactory than the design where a differential unit is used in each axle only. We do not use a locking-device unit.

C. B. Nash (Maccar Truck Co.):—With regard to a six-wheel vehicle with and without a third differential, our experience with jobs not using a third differential has been rather sad. We have had driving units fail, wheels loosen up and tires wear excessively. Naturally, there must have been some loss of power which was not noticed. Using third differentials so far, we have had no complaints; however, our experience with the third differential is only over a period of about one year.

J. Wiggers (Moreland Motor Truck Co.):—Our experience with the third differential has been somewhat limited, although we have a number of trucks running with the third differential in them. We have found that, when a truck is running on good roads and pavements, the four-wheel drive really is not necessary, this drive generally being limited to trucks that operate in sand and bad spots or over dirt roads, and the like. On trucks of this type, we have found that the third differential is really a detriment unless provision is made for a differential lock, because it allows the operator to get stuck in the sand and bad spots the same as a four-wheel truck would; therefore we would not recommend using a third differential unless a differential lock has been installed. However, we have found that, when trucks with a third differential are operating, the results to the truck seem to be less injurious, particularly when the operator starts shifting the tires around, having larger tires on one axle than he does on the other so that no uniformity exists.

C. A. Peirce (Diamond T Motor Car Co.):—We have never used a third differential in any of our trucks. Our first six-wheelers were built before the third differential in the present accepted form was available. While isolated cases of trouble may have been experienced, when proper attention has been paid to tire inflation and tire wear not enough trouble has been experienced to offset the apparent advantages of the six-wheel construction. We offer the third differential as optional equipment, but so far no customer has cared to use it, apparently preferring the increased traction which is available without it.

W. J. Baumgartner (Relay Motors Corp.):—Our opinion is that, except for very bad going, the trailer axle is the best bet, especially where you are operating on good roads.

A. C. Wollensak (Sterling Motor Truck Co.):—Our experience with conventional differentials and no third differential was confined to quite a few trucks which we built with Timken axles. The operators of these trucks, particularly in dump service, were continually in trouble. The worms and worm wheels ran very hot and the propeller-shafts were a source of constant trouble. On our own trucks, we use no other type of differential than the Krohn. If the driving tires are kept to the same diameter and the truck is operated on good roads, the third differential may not be so necessary; however, considerable doubt always exists as to which axle is driving the truck.

J. W. Shields (Firestone Tire & Rubber Co.):—I am more convinced than ever of the absolute necessity of a third differential between the two axles where power is supplied to all four rear wheels. The trouble that operators have had with units that were not equipped with the third differential has, in my opinion, been largely responsible for the unfavorable reports we have had on six-wheel operations.

J. G. Holmstrom (Kenworth Motor Truck Corp.):—We have never used a third differential, for the reason that we do not believe that the extra cost, weight and complications are worth any advantages which might be gained. Without question more four-wheel-drive six-wheel trucks without the third differential are operating in the Northwest than any other, and this type of unit is popular here because of the traction required on some of the grades and the fact that our laws are very favorable to this type of vehicle. We are getting very favorable tire mileages, from 30,000 to 60,000 miles, depending on the service.

With the four-wheel drive of the type we are using, the operator must use extreme vigilance in keeping all eight rear tires inflated evenly and the rolling radii as nearly equal as possible on the tires on each side of the truck. If this is done and the entire back-end unit is given proper lubrication and adjustment, tire wear is very good and we have no gear troubles whatever. We can detect no power loss in driving between a four-wheel-driving and a two-wheel-driving six-wheeler, although the latter coasts a little easier. If the tires are not given proper attention, the results are noticed first in tire wear and excessive pressure on the journals of the between-axle universal-joints, not particularly in the driving axles themselves, although the axles will heat up very quickly with a flat tire. Apparently, all we have to gain with a third differential is the opportunity of being more lax in watching tire size and inflation, which is not as bad as the mechanical complications, parts, service, weight and cost of the third-differential job.

G. A. Gemmer (National Motors Mfg. Co.):—Our experience with the Timken third differential has been that without this differential the jobs were satisfactory in the hands of careful operators who would watch very closely the inflation and diameter of the tires. But all operators are not careful, and with such we experienced trouble with worm wheels burning out and the like. Our experience has been such that we have made it a rule to build no six-wheeler without this third differential. We have found that as slight a difference as ½ in. in the diameter of the tires would raise the temperature in the rear axle from a satisfactory running condition to a temperature at which the worm wheel would be destroyed on a run of any length. Further, with the third differential, unless the tires are perfectly even, the job rolls much freer; that is, with uneven tires one axle will be dragging instead of helping.

Col. E. S. Stayer, U. S. A. (Holabird Quartermaster Depot):—As a multi-wheel-driven chassis has been the prime subject of extensive development, and as it is generally accorded that the addition of front-wheel-driving mechanism introduces complexity in design, efforts tending toward simplicity have been paramount. With that reasoning in mind, we looked with great disfavor on the addition of a third differential between the rear axles in a bogie. We remain adamant in this stand, unless sounder reasons for its requirement develop. In long-distance hauling, at the prevailing rates of speed, it is possibly a valuable adjunct. In military operations at controlled rates of speed, where tire replacements are as frequent as safety requires, we are positive that the third differential is not only unnecessary but is an unwarranted expense in initial cost and upkeep and, in fact, a detriment to the successful operation of the chassis.

We labored through years of discussion, similar in character, concerning the requirement of third differentials between the driving axles in four-wheel four-wheel-drive chassis—in this design, to be incorporated in the transfer case. The argument in its favor was as sound on its face as that in connection with the six-wheeler. An extensive testing program failed to disclose any sound reason for resorting to it. We further watched with interest the re-

sults of a comparable test conducted commercially, several chassis of this type being operated daily up and down a 67-mile winding grade hauling cement to one of the Western hydroelectric projects. It was the first time that balloon tires had been resorted to on commercial equipment of the four-wheel-drive type. After 33,000 miles of operation on original equipment, we believed the commercial interests were aware of the fact that it could hardly be considered a necessity.

Another branch of the War Department recently purchased a number of six-wheelers from a very reputable commercial source. They came equipped with third differentials. We have subsequently been led to understand that the department desires that the third differentials be removed as an annoyance. Commercial interests finally conceded the point that in four-wheel-drive equipment, in wheelbases over 130 in., and of not over 22-deg. front-wheel cramp, third differentials were evidently not required. Today, we are successfully operating equipment of this character in wheelbase of 106 in. and cramping angles of 32 deg. We feel that the same general conditions will eventually prevail in regard to the controversy over its use in the six-wheeler in military operations.

APPENDIX 2

Tire-Wear Experience of Operators Reported by Goodyear Tire & Rubber Co.

Company A (Milk):—The maintenance superintendent, who has perhaps had more experience in operating six-wheel-trucks than any other individual in Chicago, gave the following viewpoints:

The company operates long-distance milk trucks: seven trucks with single drive, or four-wheel type of truck; five six-wheel trucks with four-wheel worm drive, dual tires; five six-wheel trucks, single worm drive with trailing axle and dual tires and two six-wheel trucks, chain drive with trailing axle on single solid tires. In this last case, the driving-axle tires are 2 in. wider than the trailing-axle tires. The trailing axle was applied after the original purchase of the truck, the objective being to meet State load-law specifications. The types of axle used are Wisconsin and Timken; the makes of truck are Sterling, Nelson-LeMoon, White and Available.

The favored tire size is 40 x 8-in., high-pressure. The operator does not believe that he gets as much sway from this tire size as on ballon tires of corresponding carrying capacity, especially where the tires are subjected to a liquid load, as they are in this operation.

Mr. ——— decidedly favors the four-wheel-drive truck as compared with other types, for the following reasons:

(1) It gives better traction and the traction is more evenly distributed.
(2) It has four-wheel brakes.
(3) The weight is more evenly distributed on all tires.
(4) The front-drive tires do not throw dirt, stones and other debris under the rear tires as much; consequently the tires on the rear axle receive fewer cuts and punctures.
(5) Skidding action is divided more equally on the rear axle when making turns.
(6) Tires on four-wheel-drive axles will average 15,000 miles more than on a single-drive axle with a trailing axle.

Tire wear is distributed on the following basis: Left front tires will wear on an average of 25 per cent longer than right front tires, the average basis of wear being considered as 30,000 miles. Right front tires wear approximately 25 per cent longer than tires on the right rear front-drive axle and about the same as those on the left rear front-drive axle. Tires on the rear-drive axle give approximately 10 per cent less mileage than front-drive-axle tires. Trailing-axle tires average only about 30 per cent of the front-drive-axle tire mileage.

The difference in tire mileage of rear axles is not due to skidding action on turns and the like, but to cuts and other damage from debris loosened by the front tires and thrown under the rear tires. Mr. ———'s opinion is that rear tandem-axle tire mileage would equal front-drive-axle tire mileage except for the debris loosened by the front tires.

In making turns when going forward, the turning track is governed chiefly by the front-drive axle. In this case the rear axle skids. In backing up, the rear axle controls the turn and the front-drive axle does the skidding. Mr. ——— further states that, in mechanical maintenance or tire-mileage result, little or no difference has been found between the several types of rear-axle assemblies.

Tire mating is done uniformly on all positions, with about the same percentage of tread wear. Air inflation is maintained uniformly on all positions, because of the objection of the tire dealer, if, when making adjustments, he finds that tires have not been operated according to his specifications. The maintenance superintendent would prefer to operate his trucks with the inside duals carrying 5 lb. less inflation than the outside tires.

On the solid-tired trucks, on which the driving-axle tires are 2 in. wider than the trailing-axle tires, the latter are not worn out and therefore a comparison could not be made. Judging from appearances, the tires on all positions are about 40 per cent worn. Mr. ——— expressed the opinion that the addition of the trailing axle would increase the original or front-drive tire mileage about 25 per cent. He does not recommend the above experiment unless State laws require it.

Federal Motor Truck Co.:—The Federal factory representative showed me a six-wheel truck of 3½-ton rated capacity with two-wheel worm drive and a trailing axle. The truck had always carried loads in excess of its rated capacity, averaging a 5-ton payload. The 32 x 6-in. eight-ply dual tires had delivered 38,900 miles and appeared to be about 60 per cent worn, all tires showing approximately the same degree of wear.

Company B (Transportation):—The maintenance superintendent reported that the company's experience was confined to the International type of truck equipped with an Eaton axle, double-reduction herringbone-gear transmission, with two-wheel drive and one trailing axle. Tires in best condition are applied on the front wheels and on the driving-axle wheels; those in the poorest condition are applied to the trailing axle. This procedure is followed for safety reasons, as the trucks operate at high speeds. The driving axle is equipped with good tires both for safety and good traction; a flat tire on the trailing axle would not affect control of the truck.

The superintendent believes that he gets comparable mileage results from all tires on rear-axle positions. The driving-axle tires wear chiefly from brake-application traction, whereas the trailing-axle tires wear faster because of the debris thrown under them by the driving-axle tires. He does not believe that the turning and consequent slippage of rear tires materially affects tire results, as 90 per cent of his vehicle operation is confined to highway service. Tires are removed primarly because of cuts; not from wearing out. Individual-tire mileage could not be used as a criterion of position results, as tires are shifted according to the plan mentioned.

Mr. ——— favors the two-wheel drive for the following reasons:

(1) This type of truck supplies sufficient traction for his demands.
(2) He believes that he has less maintenance on this type than he would have with a four-wheel-drive truck.
(3) This type supplies additional carrying capacity as compared with the four-wheel type of truck.
(4) The six-wheel type supplies more traction than the four-wheel truck of equal carrying capacity.

Company C (Tea):—The superintendent of automotive maintenance reported that the company does not have trucks with four-wheel drive, his experience being confined to the one-axle drive and one trailing axle on Mack trucks used primarily for highway travel.

Comparatively the same mileage is secured from tires on all positions. His explanation is that, in making a turn, the driving axle controls the skid, the trailing axle actually skidding until the truck is again running straight. The braking and accelerating power on the driving-axle tires practically offsets the skidding abuse on the trailing-axle tires. The front tires, in his opinion, receive more abuse

than the rear tires, because of road crown, inertia and other factors that chiefly affect the front tires.

Mr. ——— does not believe that six-wheel trucks are practical for city operation, owing to the forced skidding of the rear tires on turns.

Mating of tires is given very little consideration, although the superintendent is thoroughly "sold" on the idea of keeping his trucks equipped with one type of tire. The average tire-mile expectancy from the six-wheel type of vehicle is 30,000 miles.

The company's chief mechanical-maintenance cost of this type of truck is in the replacement of spring-bolts, particularly in the winter, when the trucks are forced to operate over ice-rutted streets.

COMPANY D (Dairy):—The Goodyear company recently surveyed a Nelson-LeMoon four-wheel worm-drive Model HWB-100 truck of 8 to 10-ton capacity fitted with 9.75-20 dual tires. This was purchased in 1931 by a dairy company and had operated 27,860 miles on the originally equipped tires. All tires appeared to be evenly worn and none more than 40 per cent.

The truck was equipped with Westinghouse air-brakes on the rear axles only. The average load carried is approximately 7 tons and is evenly distributed over the entire body.

TRUCK AND OTHER TIRE MAKERS' OBSERVATIONS

R. T. HENDRICKSON (Hendrickson Motor Truck Co.):—From the experience we have gained in the last two years with the third differential, we feel that it has so many superior features over a straight drive that it is a necessity in a six-wheeler. For instance, in tire wear, the ——— Milk Service of Chicago has 90,000 miles on a set of tires without the expense and trouble of keeping all tires to exactly the same size and pressure. In gas mileage, Mr. ———, of Antioch, Ill., with a gross truck-and-trailer load of 72,000 lb., engine size 4¾ x 6 in., six cylinder, is getting an average of 4.25 miles per gal. With a gross truck-and-trailer load of 104,000 lb. we average about 2.75 to 3 miles per gal.

J. W. SHIELDS (Firestone Tire & Rubber Co.):—My opinion is that a six-wheel truck which does not pull a trailer should be driven on one axle only and that the addition of the extra mechanism to drive on two axles is an economic waste. Of course, if a six-wheel unit is to be driven on one axle only, the power must be applied to the last axle and not to the middle axle.

R. H. SPELMAN (B. F. Goodrich Rubber Co.):—On one of our mileage accounts a number of Safeway Coaches were run for several years. This equipment is of the underslung worm-drive type, with the two driving worms connected in tandem by a short intermediate shaft equipped with a universal-joint at each end. The driveshaft is connected to the forward end of the first worm shaft, and power is transmitted through this shaft to the second axle.

In 1930, 805 tires, size 36 x 8 in., were removed. The average mileage was 18,900 miles per tire.

Wheel Position	Tire Changes	Punctures
Left Front	184	34
Right Front	194	37
Left Center	226	21
Right Center	296	27
Left Rear	287	71
Right Rear	527	237

Quoting our engineer in charge of this operation:

> There is no differential action between front and rear differentials, which was probably the cause of the rapid tread-wear we got on the rear of Safeway Coaches. If tires were mismatched or the pressure were low in one or more tires, it would cause a dragging action. My opinion is that a number of other things were causing rapid, scuffy wear on the rear tires. Many times new tires were installed all the way round the coaches and the pressure was watched closely, but the scuffing continued. The tire mileage was low and the scuffing appeared mostly on the forward rear (center) axle. The center tires seemed to develop more heat than those in other positions, causing more bead and carcass failures, along with the rapid wear. About 70 per cent of cuts and punctures were on the right side, which is nearer the curb; this probably is due to the center tire turning the object up on end and the rear tire picking it up.
>
> Tire performance on trucks having one driving and one trailing axle results in rapid wear on the driving axle, particularly if the brakes are only on that axle.

Bearing Metals for Heavy-Duty Engines

DEMANDS for increased power have forced the development of better bearing metals, mainly along two lines. One is the refinement of tin-base babbitt metal and the other the working out of copper-lead mixtures.

High-grade babbitt metal as we know it today is susceptible to vast improvement. Ordinary babbitt metal will not suffice for an engine in hard long-distance operation of 300 to 400 miles per day with gross vehicle loads of 40,000 to 70,000 lb., no matter how carefully the rods and main bearings are cleaned and tinned and with what care and expert supervision the bearings are made. Its main fault is a tendency toward grain growth and brittleness; its ductility is lost, mainly because of the inclusion of foreign gases in the initial mixing of the metal. Higher-grade babbitts will give double or triple the mileage without losing their ductility or developing cracks and breaking down at the ends of the bearings, where the bearing loads are concentrated because of minute deflections of the shaft.

The copper-lead bearings are free from the faults mentioned; they will carry higher loads and are not weakened so seriously by high oil temperatures. The heat conductivity of the copper-lead alloys is about three times that of high-grade tin babbitt, which is another valuable quality. In desert operation in California, with atmospheric temperature at 120 deg. fahr., we have observed temperatures of 320 deg. in the oil-pan, and we know from dynamometer tests that this means a temperature of 345 deg. at the bearings.

A curve of strength versus temperature of a good grade of babbitt metal, approximating S.A.E. No. 11 in analysis, shows that its compressive strength at 70 deg. fahr. is 22,000 lb. per sq. in. and drops to 4000 lb. per sq. in. at 400 deg. The melting point of this metal was only 100 deg. higher than the known operating temperature of an engine bearing. The strength of copper-lead mixtures is about 20,000 lb. per sq. in., and its melting point of 1700 deg. is far above any possible temperature of the oil in an engine.

The technique of producing bearings of uniform quality at reasonable cost has held back the extensive use of this type of bearing metal.—From a Chicago Section paper by J. B. Fisher, M.S.A.E., chief engineer, Waukesha Motor Co.

Correction in Dickinson and Bridgeman Paper

IN the Annual Meeting paper by H. C. Dickinson and O. C. Bridgeman entitled, Fundamentals of Automotive Lubrication, which was printed in the July, 1932, issue of the S.A.E. JOURNAL, an error occurs in the fifth line of the second complete paragraph in the first column of p. 280. The last word in the line should be "larger" instead of "smaller" as printed. As corrected, the sentence will read, "As the clearance becomes larger, a lower coefficient of friction is obtained for the same value of ZN/P.

Correction in Prescott Paper

IN the paper by Ford L. Prescott, entitled Indicators as a Means of Improving Aircraft-Engine Performance, which was printed in the September, 1932, issue of the S.A.E. JOURNAL, the word "inversely" should be inserted on p. 365 in the first line of the first column. As corrected, the complete sentence, which begins on the preceding page, will read "The latter is preferable, since the rate of change of phase should be inversely proportional to the rate of change of pressure."

Judging Super-Balloon Tires

Semi-Annual Meeting Paper

By B. J. Lemon[1]

CAR engineers are haunted again by the ghost of very low-pressure tires, which they had hoped several years ago was laid. These tires are one of the elements of airplanes that they hoped not to have to adopt. As was the case with the balloon tire of 10 years ago, the super-balloon was offered by tire companies in answer to a demand of car owners for a replacement tire that would give a softer ride. The question now disturbing car engineers is whether super-balloons must be offered as regular or optional equipment on new cars.

They have an eye and sales appeal but doubts exist regarding their safety in event of blow-out, their susceptibility to puncture, their relative mileage life, their effect on steering and the like. Mr. Lemon discusses these points and the engineering changes that will be necessary in the chassis if super-balloons are to become regular equipment.

TIRE history of 1923 seems almost certain to repeat itself in 1933, for super-balloon tires confront us as a reality. The full-balloon tire, which a decade ago brought perhaps our greatest variety of automobile problems, has returned to haunt the engineers of the industry. The eye appeal of big tires tends to make us forget that wholly acceptable riding comfort can be attained only by an engineering correlation of a pillowy looking tire with a number of other interrelated automobile parts that are equally important from the viewpoints of riding and operation.

The result is that today car engineers are confronted in a correspondingly magnified degree with the perennial problems of shimmy, tramp and wheel kick and of revised steering, springing, snubbing, weight distribution and seat-cushion spring rates and deflections. These problems they must accept and try to reduce more nearly to a final solution, because the customer and the sales department will not forsake the thought of the improved ride they anticipate from super-balloon tires. And, what is most important, the customer is right; riding-comfort *can* be improved; the 1932 cars made progess over the 1931 cars in this respect. Super-balloon tires will be one of the contributing factors to further ride improvement. Therefore emergency changes are in order for the immediate future, and a more complete car redesign is predicted for the not very distant future.

The engineering reception of the super-balloon tire is as chilly as it was 10 years ago. This chilliness may be of shorter duration because of hotter competition; and the quicker and harder we wrestle with the problem, the stronger will be our courage and the sharper our skill in completing the work.

Nothing about the super-balloon tire is particularly new, except the greater degree of ballooning. Its advantages and disadvantages were listed in 1923. The justification for this paper is to review the terrors that once threatened but only partly materialized, and to form an opinion regarding the buying factors that are weighed by the car buyer who desires super-balloon tires. Last year we conjured up alarms over drop-center rims which experience proved groundless. Highway officials who viewed free-wheeling with alarm have now become excited over the assumed dangers of super-balloons. They fear harder steering, roll-off of tires on turns, sudden blowouts and other possibilities for accident increase. The extra amount of driver effort required to steer 1932-model cars fitted with larger tires is sufficient to make present steering equipment seem somewhat unsatisfactory; however, probably 50 per cent of the older cars running today steer harder than a 1932 car equipped with super-balloons. Shortcomings that are too fanciful fade into insignificance in the face of developments that sometimes force a complete readjustment of an industry.

TABLE 1—POSSIBLE SUBSTITUTIONS OF SUPER-BALLOON TIRES FOR PRESENT TIRES[a]

Present Balloon Sections, In.	(Intermediate Sections), In.	Proposed Super-Balloon Sections, In.[b]
		6.00
5.25	(5.50, 6.00)	6.50
5.50	(6.00, 6.50)	7.00
6.00	(6.50, 7.00)	7.50
6.50	(7.00, 7.50)	8.25
7.00	(7.50, 8.25)	9.00
7.50	(8.25, 9.00)	9.75

[a] All sizes in this table are nominal.
[b] The proposed sections of super-balloons are triple oversize for present balloon sections.

When Is a Tire a Super-Balloon?

Until the present chaos is restored to order, no clear definition of a super-balloon tire is possible. Original-equipment tires now include six sections, from 5.25 to 7.50 in. Super-balloons are proposed in seven sections, from 6.00 to 9.75 in. Assuming that the 6.00-in. super-balloon is intended for future light cars, possible direct size-for-size substitutions of proposed tires for present tires are shown in Table 1. The proposed tires are triple-oversize sections on smaller-diameter wheels.

Proposed experimental sizes of super-balloon tires, including nominal sections, rim widths, actual section-widths and tire standing-heights, are shown in Table 2, adopted by the Tire & Rim Association on May 13, 1932.

TABLE 2—TIRE AND RIM COMBINATIONS AND DIMENSIONAL LIMITS OF LOW-PRESSURE BALLOON TIRES FOR PASSENGER-CARS, EXPERIMENTAL PRACTICE

Nominal Tire Size, In.	Rim Size, Low-Pressure Rims Only	Maximum Tire Dimensions: Nominal Rim Diameter, In.[a]	Maximum Tire Dimensions: Tire-Section, Width, In.	Maximum Tire Dimensions: Tire Over-All Diameter, In.[a]
6.00	4.00 D	16	6.30	28.42
6.50	4.50 D	16	6.80	29.18
7.00	5.00 E	16	7.40	30.24
7.50	5.50 E	15	8.05	30.16
8.25	6.00 E	15	8.60	30.86
9.00	6.50 F	15	9.30	31.92
9.75	7.00 F	15	10.00	32.78

[a] Other rims have proportional tire heights.

[1] M.S.A.E.—Field engineer, tire-development department, United States Rubber Co., Detroit.

TABLE 3—LOADS AND INFLATION FOR LOW-PRESSURE BALLOON TIRES FOR PASSENGER-CARS, EXPERIMENTAL PRACTICE

Minimum Inflation-Pressure, Lb. per Sq. In.	6.00		6.50		7.00		7.50		8.25			9.00			9.75	
	15	16	15	16	15	16	15	16	15	16	17	15	16	17	15	16
16	560	585	665	695	770	805	875	915	985	1,025	1,070	1,090	1,135	1,185	1,195	1,245
17	595	620	705	735	815	850	925	965	1,040	1,080	1,130	1,150	1,195	1,245	1,260	1,310
18	630	660	745	780	860	900	975	1,020	1,095	1,140	1,190	1,210	1,260	1,310	1,330	1,380
19	665	695	785	820	905	945	1,025	2,070	1,150	1,195	1,250	1,270	1,320	1,375	1,395	1,445
20	700	735	825	865	950	995	1,075	1,125	*1,205*	*1,255*	*1,310*	1,330	1,385	1,440	1,460	1,515
21	735	770	865	905	995	1.040	1,125	1,175	1,260	1,310	1,370	1,390	1,445	1,505	1,525	1,580
22	*775*	*810*	*905*	*950*	*1,040*	*1,090*	*1,175*	*1,230*	1,315	1,370	1,430	1,450	1,510	1,570	1,590	1,650
23									1,370	1,425	1,490	1,510	1,570	1,635	1,655	1,715
24									**1.425**	**1,485**	**1,550**	**1,570**	**1,635**	**1,700**	**1,720**	**1,785**

Note.—Italic figures indicate maximum loads for four-ply tires; bold face, for six-ply tires.

Unfortunately, a size-for-size replacement triple-oversize tire is not practicable on all present cars. Cars using 7.00 and 7.50-in. balloon tires generally have 15-in. brake-drums, which need 17-in. rims for satisfactory clearance. Therefore the 9.00-16 and 9.75-16-in. super-balloons, if used on a 16-in.-diameter wheel, will require brake-diameter changes. Because of their height and width, these two sections cannot replace present tires on 17-in. wheels without some fender, body or probably steering hook-up changes and without a possible sacrifice in tire appearance.

Further confusion is thrown into the table and definition by present field replacement practice. Tire makers began supplying a 9.00-in. super-balloon for Chevrolet, Ford and Plymouth cars. This is a septuple oversize for the 5.25-in. standard balloon. However, saner ideas prevailed and we dropped to the 7.50-in. super-balloon, which is a quintuple oversize; and real conservatism asserted itself when the 6.50-in. super-balloon, a triple oversize, was offered for field replacement. Car engineers have kept extreme over-sizing well in check. The Graham-Paige Motors Corp. chose the 7.50-in. size, a triple oversize; and the DeSoto and Rockne cars are offered with quadruple oversizes.

Loads, Inflation and Tire Mileage

Merely selecting a certain oversize does not give a large-section tire a super-ballooning ride-effect unless low enough pressures are used to compensate in cushioning for the increased tire size. The tire industry has worked out and agreed upon Table 3 of loads and minimum-inflation pressures, based upon approximately 18-per-cent tire deflection. This table, which was adopted as experimental practice by the Tire & Rim Association on May 13, 1932, is based on the assumption that the tires will give best service if run with the indicated loads and inflations and is believed to be a reasonable set-up, keeping in mind, first, tire service and a stable car, and, second, the desire for some improvement in the automobile ride. A softer ride is possible at lower than the recommended air pressures but at some sacrifice of tire life; a more stable car will result from higher pressures than those recommended but riding comfort will be decreased.

Development of the super-balloon tire has not yet progressed sufficiently to claim that tire-tread life will exceed that of the present standard tire. Fig. 1 shows superimposed, for comparison, the tread areas of the 6.00-17 four-ply standard tire and the 7.50-15 four-ply super-balloon tire at the maximum recommended load for the 6.00-17 tire; that is, 1025 lb. This increase in area of 50 to 65 per cent for the super-balloon leads one to expect longer tread wear; but rate of tread wear depends largely upon the amount of scrubbing action that the tread undergoes, and the rate of this action increases with increased area when caused by lower inflation-pressures or higher loads.

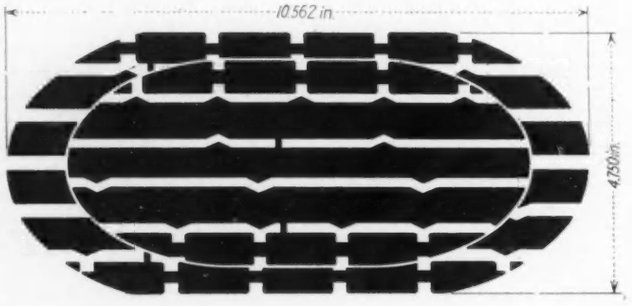


FIG. 1—COMPARISON OF TREAD-CONTACT AREA OF 6.00-17 BALLOON AND 7.50-15 SUPER-BALLOON TIRES, BOTH FOUR-PLY, WITH A LOAD OF 1025 LB.

The Gross Contact Area of the Super-Balloon Is 63 Per Cent Greater than That of the Balloon Tire and the Net Contact Area Is 53 Per Cent Greater

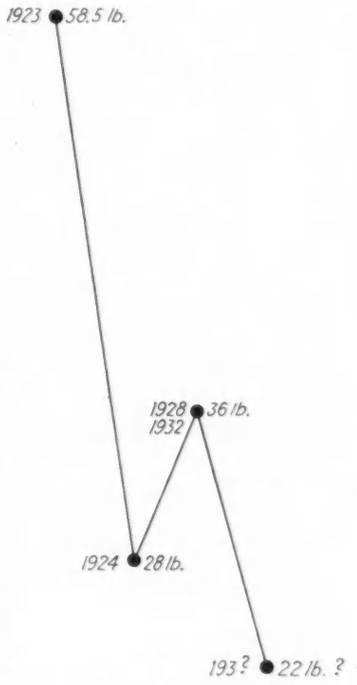


FIG. 2—TEN-YEAR TREND OF INFLATION PRESSURE

Pressures Shown Are Averages of Recommendations of All Car Makers for Front and Rear Tires for Years Indicated

Tests run with inflations that allow a tire deflection of 22 per cent show an abnormal rate of tread wear. Some tests at 18-per-cent deflection indicate that the super-tire gives tread service equal to that of the corresponding standard tire. To maintain accurately any set of deflection of the super-balloon, such as 18 per cent, will be difficult because a loss of 1 lb. of air pressure at 22-lb. inflation is a much higher proportion than at the 32-lb. pressure commonly used in the standard balloon tires, which means a relatively more rapid increase in deflection of the super-tire and a consequent increase in tread wear.

Because of increased deflection, tire heating causes a more rapid increase in air pressure in super-balloons, up to a temperature at which the radiation of heat from the larger exposed tire-surface produces an equilibrium of temperature and pressure. The fact that temperature and resulting inflation-rise in the super-balloon are rather rapid means that tire pressures, from the viewpoints of riding quality and tire life, will not be easy to

control. This control, with respect to air leakage and the tread wear closely allied to it, must and will be given early attention by tire engineers. No tire engineer doubts that tread life of super-balloons eventually will approximate that of standard balloons.

Fig. 2 shows that the proposed reduction in air pressure is by no means so radical as occurred in 1923. The relative increase in tire sections of proposed super-balloons over balloons and high-pressure tires is indicated graphically in Fig. 3.

The automobile of today is more adaptable to super-balloons than the car of 1923 was to the balloon tire. This is shown by the almost overnight application by several car makers, with relatively hasty and minor alterations, of the larger tires as optional equipment.

Effect on Car Appearance

Increase in car speeds, requiring more stability and a lower center of gravity of the car, has caused a gradual reduction of wheel diameters from 23 in. to 17, the increase in tire sections compensating for only about one-third of this axle drop. Body lines and sweeping fenders to accentuate the speed aspect have heretofore not been seriously affected by the increases in tire sections. Today, cars are so close to the low limit of road clearance that further wheel-diameter reduction must be offset largely by correspondingly larger tire sections, such as those of the triple-oversize super-balloons, which rather radically change the appearance of the car and tend to increase the stability but to decrease the speed aspect. Whether present car lines are maintained or a semi-teardrop design becomes the vogue, triple-oversize tires, if adopted, will compel extensive changes in fenders, both for appearance and for accommodating the tires in the fender wells. Some 1932 cars equipped with super-balloon tires are shown in Fig. 4.

Super-Balloon Design a Step in Evolution

The development of super-balloon tires and their use on automobiles and airplanes date back several years. As a rule, the super-tire follows the conventional design of its balloon predecessor, an exception being the streamline type, the practicability of which time will determine.

Strength of the carcass fabric in the super-tire is calculated, as heretofore, on the load to be carried. The larger cross-section reduces localized strains by distributing them over larger areas. Tread design with respect to width, non-skid depth and configuration, while not entirely settled, is in proportion to the increased size of the tire and the expected increase in tread wear from the greater scrubbing action of the larger area of the tread in contact with the road. The rubber in the carcass and on the tread and side wall is as strong as that in any other type of tire. The design of the beads and the wire strength are in proportion to the rim width and the tire loading. Rims are proportionately much wider than heretofore, as will be seen in Fig. 5, which makes for stability of car performance with both inflated and flat tires.

With the advent of free-wheeling, vacuum-operated clutches, quieter transmissions and easier gearshifting, the physical exertion of driving was reduced and riding became more comfortable and pleasurable. Whether super-balloon tires ever are widely adopted and are per-

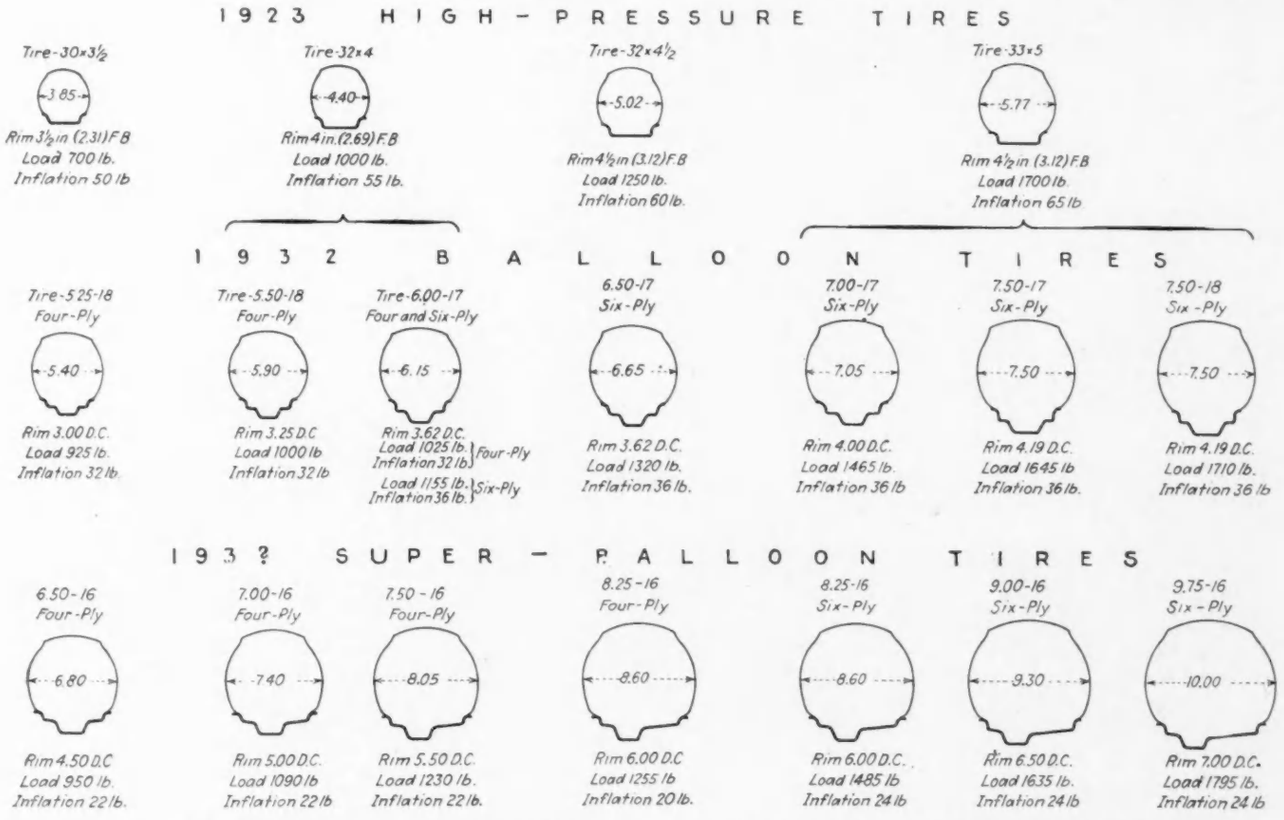


FIG. 3—COMPARISON OF SECTIONS OF 1923 HIGH-PRESSURE, 1932 BALLOON AND PROPOSED SUPER-BALLOON TIRES

Rim Diameters Are Indicated Only for Illustrative Purposes. The Average Increase of Balloon-Tire Section over That of High-Pressure Tires Is 31 Per Cent; That of Super-Balloons over Balloons Is 30 Per Cent. Width of Rims for Super-Balloons Averages 55 Per Cent Greater than for Standard Balloons

manently retained or are a passing phase is of less moment than that the public be given a better ride, no matter by what methods. They will stimulate work on cushions, springs, shock-absorbers, unsprung axle weight and independently sprung wheels, along with other problems involved in front ends that get nervous when air pressure is neglected.

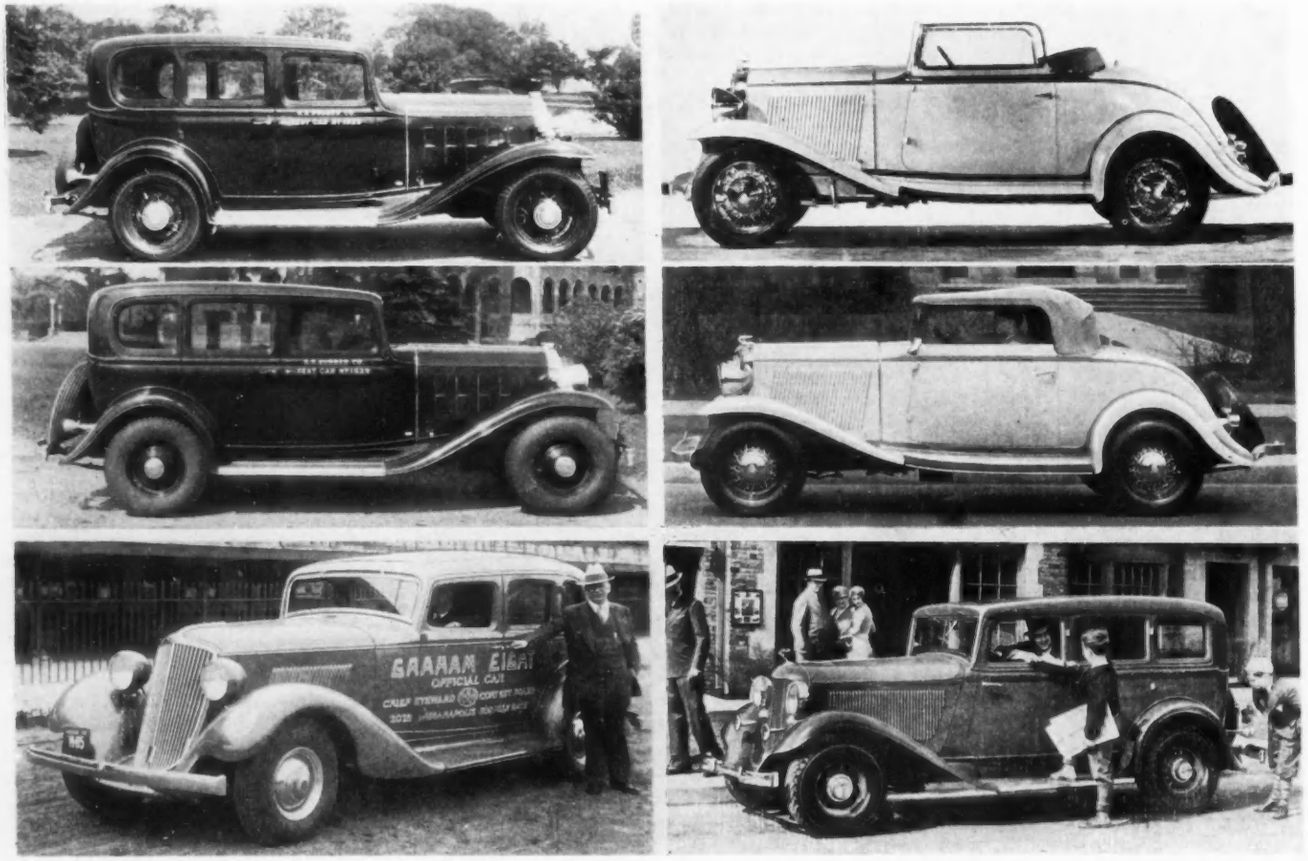
The portal of entry of the balloon tire in 1923 was by the side door, as a replacement tire in the field. The super-balloon got its impetus in the same way. Even in present hard times, the public seems willing to part with $50 to $100 to have something different in tire equipment, without adequate assurance by either the vehicle or the tire maker that the car as a whole will perform better or even as well as before. The new tire has eye appeal as well as a certain popular ride appeal. Replacement sales started with the larger distributors in the larger cities where rough streets permitted of convincing demonstrations, and the demand exceeded expectations to so great an extent that car dealers passed back the word to their general sales managers that super-tires should be offered as optional equipment.

The engineers balked at this idea. They had just released new models, of the best value and performance that had ever been offered to the customer, and were thinking, perhaps, about body streamlining and independently sprung wheels, hence did not relish another major change in tire type. Tires are giving phenomenal service, and the cars are designed to perform on medium air pressures; therefore, why disorganize design when curtailment of development expense was the order from the comptroller?

Had the car manufacturers in the National Automobile Chamber of Commerce and the equipment-tire makers recognized the magnitude of the changes that super-balloons may eventually require and the immediate expense of the necessary development work, the movement might have been retarded and adequate time allowed, as in 1923, to work the tire into the design of new cars. Now it is too late for any retardation pressure to become effective. Super-balloons very likely will be standard equipment on some light and medium-weight cars in 1933, unless serious deficiencies that are not now apparent develop.

I believe that car buyers do not follow a new movement, such as super-balloon tires, like sheep going through a fence. Those who buy blindly and are dissatisfied complain enough to expose quickly an over-advertised innovation. No general dissatisfaction with super-balloon tires seems to have developed as yet, in spite of some of the monstrosities perpetrated in the change-over. Some sensation that is pleasing to the car occupants is afforded by the ride on super-balloons that has been attained in no other way on cars of present design. The super-balloon is another step in tire evolution.

Table 4 indicates that, size for size, super-balloon



(Upper Left) Buick Model 60 Equipped with 6.00 Standard Balloons and (Center) with 8.25 Super-Balloons
(Lower Left) Graham 8 Fitted with 7.50-15 Super-Balloons Offered as Optional Factory Equipment
(Upper Right) Rockne 65 Equipped with 5.25-18 Standard Balloons and (Center) with 7.00-16 Super-Balloons Offered as Optional Factory Equipment
(Lower Right) DeSoto Equipped with 7.50-15 Super-Balloon Tires

FIG. 4—EFFECT OF SUPER-BALLOON TIRES ON THE APPEARANCE OF CARS

TABLE 4—TIRE AND WHEEL WEIGHTS, EXPERIMENTAL PRACTICE

Casing and Tube						Wheels				Tires and Wheels	
1932 Standard Balloon			1932 Super-Balloon			1932 Standard Wheel		1932 Super-Wheel		1932 Standard,	1932 Super,
Size	Plies	Lb.	Size	Plies	Lb.	Rim Size	Lb.	Rim Size	Lb.	Lb.	Lb.
5.25-18	4	20.5	6.50-16	4	24.0	18×3.25 D.C.	25	16×4.50 D.C.	22	45.5	46.0
5.50-17	4	22.9	7.00-16	4	27.7	18×3.25 D.C.	25	16×5.00 D.C.	24	47.9	51.7
6.00-17	4	25.3	7.50-15	4	29.0	17×3.62 D.C.	27	15×5.50 D.C.	25	52.3	54.0
6.50-17	6	35.4	8.25-16	6	37.2	17×3.62 D.C.	27	16×6.00 D.C.	27	62.4	64.2
7.00-17	6	42.2	9.00-16	6	44.0	17×4.00 D.C.	30	16×6.50 D.C.	30	72.2	74.0
7.50-17	6	48.8	9.75-16	6	49.0	17×4.19 D.C.	35	16×6.50 D.C.	34	83.8	83.0

casings, tubes and wheels as built today are slightly heavier than standard casings, tubes and wheels. The wheel weights given are an average of different types of equipment, and the tires are still in a more or less experimental stage with respect to weight.

Deceleration, Maximum Speed and Gasoline Economy

The rolling resistance of super-balloons, as measured by deceleration, maximum speed and the gasoline economy, seems to be slightly greater for the larger tires than for standard balloons. A series of tests on a light car gave the following results:

Time required to coast from 30 to 20 m.p.h. with standard 5.25-18 four-ply tires inflated to 32 lb., 14.9 sec. The same car fitted with 7.50-15 four-ply super-balloon tires inflated to 20, 15 and 12 lb., required 17.3, 16.4 and 15.6 sec. respectively.

Maximum speed, with 5.25-18 four-ply standard tires, inflated to 32 lb., was 63.8 m.p.h.; with 7.50-15 four-ply super-balloons at 15 lb. air, 64.1 m.p.h.; and with 9.00-13 tires and 15 lb. of air, 64.9 m.p.h.

Gasoline consumption with four different sizes of tire, inflated to Tire & Rim Association recommendations, was at the rate of 1 gal. per 13.6, 13.3, 13.2 and 13.4 miles for 5.25-18, 6.50-16, 7.00-16 and 7.50-15 tire sizes respectively. The consumption of a heavier car, using 6.00-17 four-ply standard tires at 32-lb. inflation and 7.50-15 four-ply super-tires at 22-lb. inflation, favored the standard tire by 0.2 mile per gal. at 25 m.p.h. and 0.35 mile per gal. at 45 m.p.h.

All of these tests were made on flat roads. Results of tests on rolling highways may tell a different story.

Shimmy and Tramp

We know more about shimmy and tramp than we did 10 years ago. They converted the balloon tire into a medium-pressure tire and robbed us of the kind of ride we are now striving for in the super-balloon. When super-balloons were tried on a number of cars two years ago, front-end instability caused them to be discarded before the sales departments became conscious of their eye and cushioning appeal. Now engineers are in a different mood; they can do things they could not do a few years ago, because they have to.

Every car on which the new tires have been tried has developed front-end nervousness; some considerable, others very little; but corrective measures, either by car engineers or some of our expert service-station mechanics, have in every instance in my experience produced a remedy that quieted the tramp and shimmy at least for a time and produced a reasonably acceptable ride at rather low air-pressures. We could not have done this 10 years ago.

Why is this? First, I believe that we are becoming more openminded and resourceful in trying new ideas. We should and do know more about the causes of tramp and shimmy; the effect on them of changes of caster, spring stiffness and unsprung weight; and about lateral and radial tire and wheel run-out and wheel and tire unbalance. Instead of condemning the super-tire, we are trying systematically to find out what is wrong with the car as well as with the tire and wheel.

With this improved attitude and better knowledge of engineering and service, it is not beyond belief that most present-model cars can be doctored to perform reasonably well with super-balloons and that new cars can be designed to perform on them as well as or better than with present standard tires. Tests have shown that shock-absorbers need more static control and that shock valves with smaller or differently located orifices have reduced front-end bobbing; while tighter and stronger steering-rod connections, wider front-wheel tread, reduced caster, less spring lubrication, wider rims and less wheel run-out are some of the factors that may help remove tramp and shimmy excited by the super-balloons.

Changing design to remedy one condition often produces trouble somewhere else in the car. The problem calls for systematic development of the highest order if larger tires are to be offered as original equipment. Super-balloons will reveal some of the rough spots in design, the existence of which was not advertised by the medium-pressure tire.

Rideability and Roadability

The better ride which the super-balloon affords is said by some to be confined to service on good and bad city streets, such as taxicabs encounter, or on rough country roads. The experience of test drivers and a limited number of car owners who drive on super-balloons for long distances is that a less fatiguing ride is

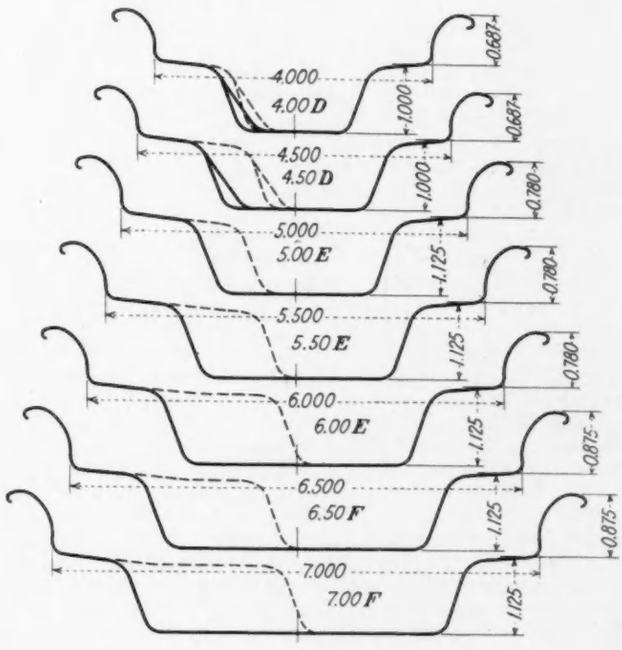


FIG. 5—DROP-CENTER RIMS FOR LOW-PRESSURE BALLOON TIRES

Adopted as Experimental Practice by the Tire & Rim Association, April 15, 1932. These Rims Have an Average Width 55 Per Cent Greater than Present Standard Drop-Center Rims

afforded also on smooth highways. Interesting information might be obtained if Dr. F. A. Moss would check, with the wabblemeter, driver fatigue after riding 300 miles on super-balloons as compared with standard tires. Innumerable shocks and vibrations, both large and small, which on short rides do not seem to cause weariness, may, if continued for several hours, be a large factor in driving fatigue. Test drivers using the larger tires say that they are fresher after a day's run on super-tires than with the standard type of tire, and some test drivers are rather good judges of riding-quality of cars as regards fatigue.

No one would expect all the factors entering into the roadability of 1932 cars to be as well coordinated with super-tires as with standard tires. Car roadability is worked out around tire size and inflation. Emergency changes to improve front-end instability and bobbing have caused cars to wander on the road and have sacrificed other desirable elements in the handling and roadability. Side roll, the amount of which depends upon tire section, inflation and rim width, gives a slightly uneasy feeling at first, but this soon wears off, for, instead of receiving a sharp side jolt that makes us reach for a door handle, we experience a sway of a decidedly softened type.

Roadability of each make of car is a distinctive property that the engineer has painstakingly worked out, but no forward-thinking person doubts that roadability can and will be improved when the engineers seriously begin designing a car for super-balloon tires.

How Super-Balloons May Affect Steering

Steering is inherently harder with the larger-section low-pressure tires. Whenever a loaded wheel fitted with a pneumatic tire is turned on its vertical axis, all of the flat part of the tire in contact with the road slides except a small area in the center of contact, and mathematically that area is a point. Pneumatic tires under load are flat at contact, to a degree depending upon the stiffness of the carcass, the load and the inflation. As super-balloons have a greater contact area than have balloon tires, more steering force is required to turn them on the steering-knuckle pivots. But saying that large-section tires steer harder than tires of smaller section will not stop people from wanting both a comfortable ride and an easily handled car, even if we seem to have gone to the limit of our ingenuity and finances in our efforts to solve the steering problem with present equipment.

Steering ratios of manually operated equipment are reliably stated to have about reached the limit of a driver's ability to spin the steering-wheel quickly for a sharp turn at high speed. Ratios vary from 12:1 to 20:1 over the price range of present cars, the higher priced automobiles using the higher ratios. The limiting factor, therefore, for 75 per cent of our automobiles seems to be price and not ratios. We can make many of our light and medium-weight cars steer easier by increasing the steering-gear reduction, at what should be a small or possibly no increase in cost based upon larger production.

For the largest and speediest cars that are near the reduction limit, the problem is one of either power steering, which is likely to be more complicated and decidedly more expensive, or using a more efficient type of manually operated gear. Considerable valuable work is being done on power steering, which should be presented soon before the Society. Non-friction-bearing types of steering equipment are in use, the efficiency of which is shown to approach double that of older types of equipment. A roller-bearing gear was installed in one of our light test cars equipped with triple-oversize super-balloon tires, and results bear out all the claims of the gear manufacturer. Eventually, cars fitted with super-balloon tires will be made to steer easier than the cars fitted with present tires, for progress is the keynote of the automobile industry.

What May Happen Following Blowouts

Whenever a new method of attaching the tire to the wheel is devised, or when a larger-section casing is proposed for use, the blowout bugaboo looms large in our imagination. Last year the drop-center rim was the cause of our fears; this year it is the super-balloon tire, which by some is believed dangerous for high-speed driving.

The chances of instantaneous tire blowouts under normal operating conditions are remote, but when a sudden blowout occurs at a speed of 50 m.p.h. or more, serious consequences may result to an inexperienced driver regardless of the type of the tire.

A series of blowout tests were run recently to compare driving control with 7.50-15 four-ply tires on 5.00-in. drop-center rims and 6.00-17 four-ply tires on 3.62-in. drop-center rims. Holes were blown in the tire and tube side-walls by cartridges electrically discharged by a contact switch in the car. The speed of the car, which weighed 3800 lb., was between 65 and 70 m.p.h. when blowout occurred. The tires went flat, as in Fig. 6, in approximately 1 sec. The experience of the driver was that the super-balloon tire, when blown on a front-wheel, produced a harder pull on the steering-wheel toward the side of the blowout than when the standard size tire let go and that it might be difficult for the average driver to right the car. However, after the steering had been righted, the car control, down to a stop, seemed easier with the larger tire than with the smaller one. This probably is due to greater rim width and to less "fight" in the more flexible, larger-section tire on the wider rim. No appreciable difference in car handling was observed when blowout of the two types of tire occurred on rear wheels.

Fortunately, blowouts are of infrequent occurrence, regardless of the type of tire. Experienced drivers know, and the inexperienced should commit to memory forever, that when a sudden blowout happens at high speed, first thought must be given to steering rather than to brake application. When the car is again under steering control, after the first shock, the brakes should be cautiously applied, intermittently on and off, until the speed is reduced.

Cars seem to be more difficult to control when a rear tire blows than when a front tire blows. In spite of the hard pull on the steering-wheel when a front tire goes down, the driver has definite directional control of steering. He has no such definite control of the rear end of the car with a flat rear tire, and rear-end sway at high speed is controlled with difficulty.

I know of no blowout tests of 8.25 or 9.00-in. tire sections. My belief is that the larger sections, with the correspondingly wider rims, may offer some increased difficulties of car control when the tire suddenly goes down. The wider rims will have a steadying effect on the car, but the effort required on the steering-wheel to right the car may be greater. Also, with the wider rims the flat tire will fold more evenly on the rim at road contact, at least at speeds above 35 m.p.h., with less flapping and fighting than is experienced with the present stiffer tire on narrow rims.

When the bugaboo of blowout is raised, and cold chills run up and down one's back, it is reassuring on inquiry to find generally that neither the skeptic, unless he is a racing driver, nor any of his friends or acquaintances has been through an experience of a sudden blowout at high speed with a modern tire. However, no rule is so general that it admits of no exception.

A few months of experience with super-balloon tires on automobiles is not sufficient to warrant conclusions that these tires are or are not more susceptible to punctures than balloon tires are. A theory advanced in favor of the balloon tire in 1924, that the lower the air pressure is, the less the tire is susceptible to punctures, still remains to be disproved. The contention that a nail or other puncturing object is less likely to be started and driven into a low-pressure tire than into a medium or high-pressure tire may have some merit, but I know of no series of tests that establish the facts.

The balloon tire covered more ground-contact area than its predecessor, the high-pressure tire, and therefore should come into contact with more puncturing objects during its life; however, I believe that punctures have not increased with the use of balloon tires. The super-balloon will cover considerably more ground than the balloon tire does, but I look for no increase in the puncture hazard, although experience may establish some facts within a year to disprove this belief. Experience on our test cars thus far indicates a reduction of punctures with super-balloons.

When the average speed of cars was lower than it is today, slowing to a stop when a tire was going or had gone flat as a result of puncture was accomplished in a relatively short distance and proportionately less damage to the tube occurred. Higher speeds and more wheel revolutions per mile because of smaller diameters are causing greater tube damage today than ever before. The drop-center rim has further aggravated tube damage, since the tire is more likely to creep on the rim, carrying the tube with it and usually pulling the valve from the tube when the valve-stem is locked to the rim. Valve-stems that pull out of the rim reduce tube damage when tires are run flat.

Punctures and tire trouble in general occur so infrequently nowadays that most drivers become careless about stopping to inspect their tires, even after car handling indicates that the air in a tire must be far below the normal pressure.

Tires Becoming More Quiet

Contact of the tire-trend configuration with a smooth road surface produces a sound of a fairly constant vibration frequency that is transmitted to the ear by the surrounding air. The tone of this class of tire sounds seems, in general, to be lower for super-balloons than for standard tires and therefore to be less objectionable.

Tires produce, inside the car, a drumming sound that is set up at the interface of the tire and the road and transmitted to the rider's ear and body through the media of the wheels, chassis and the car body. This sound, when loudest, takes on characteristics of the continuous but not very disturbing roar or rumble of a subway train as heard and felt on the street above the tunnel. It is deepest and loudest on gravel, brick and similar types of rather rough hard-surface roads and increases, apparently, with the volume of air and the sectional size of the tire.

Perfectly smooth tires on perfectly smooth roads should develop no sounds or rumble of any kind. We have today neither of these prerequisites to a noiseless tire. Highway surface has improved greatly in smoothness within recent years, but for reasons of traction and the effect on the road surface of moisture and temperature, roads never will stay perfectly smooth.

It is doubtful if tires ever again will have plain treads. Tread design has changed decidedly from smooth-tread days, as well as from the days of large knobs required for traction on dirt roads, to small configurations and to ribs with non-skid shoulders which are better suited to improved road surfaces from the viewpoint of both tire wear and tire sounds. Progress has been made in producing disorder in the vibration frequency of non-skid tread sounds, by changing the relative size and length of the tread configurations. The disorder and sound-pitch reduction thus produced do not obtain throughout the life of the non-skid, and the result is a recurrence of tire sounds at certain periods. But the progress that has been made spurs us on to greater things.

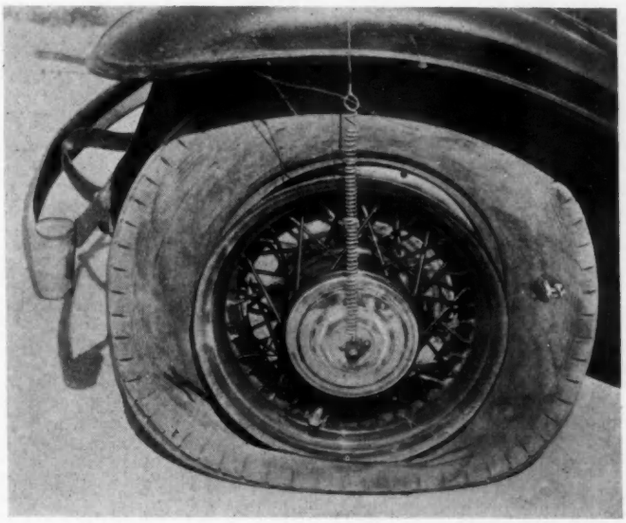


FIG. 6—A 7.50-15 FOUR-PLY SUPER-BALLOON ON A 5.00-IN. DROP-CENTER RIM BLOWN OUT AT 65 M.P.H. AND RUN FLAT 500 FT.

Pull on the Steering-Wheel Was Harder but Car Control While Decelerating Seemed Less Difficult than When a 6.00-17 Balloon Tire Was Blown

Since tire sounds are closely related to the amount, size, shape and position of the non-skid configuration on the tread, the continued need for rough-surfaced treads for present driving is a much more open question today than it was five years ago. Car engineers are rightly much more sensitive to tire sounds outside the car and tire drumming inside the car than is the public. The public seems to like plenty of slashes in the tread rubber, believing that they make for safety; but the car owners get along just as well with tires that have a combination rib and non-skid tread design. Some interesting results developed from straightaway traction and side-skid tests with current tread designs help to throw light upon the exaggerated value of and the need for extensive tread configurations as viewed from the standpoint of traction.

Traction and Skidding Resistance Improved

Because super-balloons absorb more of the road inequalities than do standard tires, traction and resistance to skidding are improved. These improvements are of special interest to the highway engineer, the vehicle designer and the tire maker who is asked to supply tires that will offer a resistance to slipping which is as high as possible and also independent of the speed and surface conditions.

As regards straightforward traction, the super-balloon excels the balloon tire because of the greater contact area and the lower air pressure. It is almost impossible to slip the super-balloon on a dry or a clean, wet surface. Because of this, some of the tire wear and tear occasioned in the past by spinning or sliding the wheels will be transferred to other car parts.

Again, the tires are more resistant to side skidding, which occurs on cornering or on steeply cambered surfaces when the wheels continue to revolve but are inclined to slip at right angles to the direction of rolling. Some past experiments have shown that large tire-contact areas are to be avoided when high resistance to sidewise force is desired. Tests made thus far with super-balloons do not coincide with this conclusion. The larger tire-tread contact area permits the tread to cover a greater number of road-surface irregularities. The lower unit pressure reduces the likelihood of the tire bouncing sidewise and thus losing maximum frictional contact with the road surface. The slight roll of the tire reduces the magnitude of the initial side thrust on the wheels, thus preventing the start of the skid.

The advantages of large-section tires for the landing and taking-off of airplanes on sandy, muddy or snow-covered fields are applicable as well to the automobile operating under similar surface conditions. A good old-fashioned winter's experience will be needed to prove that super-balloons can operate successfully everywhere without chains, but certainly fewer chains will be used for super-balloons than for the present balloon tire. From the viewpoints of traction and skidding, the new tire should prove relatively superior to the present tire under any operating conditions.

If super-balloon tires are judged solely on the basis of the advantages and disadvantages that accrue from their use on 1932 cars, they stand condemned today by an almost unanimous vote of the automobile engineering fraternity. Their advantages, generally acknowledged, are better traction, questionable appearance appeal and questionable ride appeal, all powerful factors, however, influencing a transportation-buying public. Their disadvantages include a greatly complicated and exceedingly expensive car and a wheel and tire-development program thrust upon the automobile and allied industries at a time when economic survival hangs in the balance. If the super-balloon is just another sensational gadget to attach to 1932 cars to attract a relatively small percentage of additional sales, it should and will be shown the door quickly. But if, as I believe, larger-section tires for smaller wheels are still part of the evolutionary tire movement that has been under way since buggy days, then these new tires, whether double, triple or quadruple oversizes, must be reckoned with seriously in future automobile design.

THE DISCUSSION

J. H. HUNT[2]:—Fig. 2 of the paper shows a 39-per cent decrease in tire pressure from 1923 until the balloon-tire situation became stabilized in 1928. It also shows a predicted 39-per cent decrease between the present tire pressure and that proposed for the super-balloon tire. Expressed as a percentage, the proposed change is as great as the one that caused so much trouble between 1923 and 1927. I hope the tire manufacturers, for the sake of completing the record, will include with this paper curve sheets showing the variation in the "rate" of the tire, considered as a spring, and also the hysteresis loss for a deflection cycle for the 1923 tires, the 1932 balloons and the proposed super-balloons. Such data will be of some assistance in estimating the effect of changing an important element in the very complicated dynamic system in a moving motor-car.

Mr. Lemon has suggested some of the changes that might assist in handling the super-balloons. When balloon tires appeared in 1923, we had "in the bank" certain possible changes in steering-gear. From the available balance we have already expended in many cases more in the way of reduced friction in the steering system than is really desirable, and in the case of our larger cars we seem to have used everything in the way of increased ratio between wheel and steering that is consistent with safety.

The law of diminishing returns applies to improving riding with increased tire sections, as in everythng else. The car engineer is justified in being very careful in analyzing the benefits that may appear during development and balancing these against the cost. In doing this, he should take due notice of the benefits of the same added expenditure in other units.

The suggestion has been made that the real advantage of the super-balloon tire comes from reduced body rattles. Possibly a smaller use of rubber in a more economical form than on a tire would do more to reduce body rattles than the super-balloon tire can accomplish.

J. C. TUTTLE[3]:—We have learned considerable about these low-pressure tires in the last four years of experimental work, but we still have considerable to learn. A large amount of our knowledge is coming from the car engineer, and, as with every new article, the greatest amount will come from the public.

Shimmy is not a factor in all cars. I disagree with Mr. Lemon's statement on this point. We have had a large number of applications in which there is not the slightest evidence of front-end nervousness. We have also had a number of change-over applications that have defied every known remedy for the trouble.

Mr. Lemon states that the public believes in non-skid tread designs but gets along as well with rib-tread tires. We know from our tests that the broken-up design has a definite value as regards both skidding and traction. It is also indicated that this is as true of low-pressure tires as of the present type.

Some peculiar tread wear occurs on these low-pressure tires. Scrubbing on either side of the center of the front tires and on the inside of the center of rear tires is one of the big problems in tread wear. This problem is not confined entirely to the tire, and we probably shall learn something on this point from the car engineers.

From a commercial standpoint I think we can safely say there is definite sales value in the improved riding-comfort and appearance. The value of improved appearance can best be appreciated by seeing two cars of one make and style set up together, with low-pressure tires on one and standard tires on the other. No question arises in my mind that the large tires make a great improvement and that the car with the standard tire appears out of date.

A field exists for these tires, especially on the small, light cars. Drive for a week on these large tires and then go back to the standard tires, and I am sure you will appreciate the greatly improved ride.

Tire and Shock-Absorber Effect on Body Oscillation

TORE FRANZEN[4]:—This discussion treats only of riding-quality. As riding-quality is affected by numerous details other than tires, we can touch only on the func-

[2] M.S.A.E.—Patent section, General Motors Corp., Detroit.

[3] M.S.A.E.—Development engineer, Goodyear Tire & Rubber Co., Akron, Ohio.

[4] M.S.A.E.—Experimental engineer, engineering division, Chrysler Corp., Highland Park, Mich.

tioning of the major units such as springs and shock-absorbers. In general, we have been able to produce acceptable riding-qualities in our cars, and if the public demands large tires we shall not object.

It is safe to say that low spring-periodicity is essential and such matters as lubrication must not cause trouble, as we would then deal with too sensitive a factor to be practical. In our own experience we have used conventional springs mostly but have also employed polished springs with Oilite inserts between the leaves and applied copious lubrication with good results.

It appears to be desirable to change the characteristics of shock-absorbers. However, it is too early to predict the exact type of shock-absorber control that will be generally adopted, but since Mr. Lemon has referred to static-valve shock-absorbers, which are used by Chrysler Motors in connection with air wheels, we shall give our reason for their adoption. By the trial method we have found that it is possible to obtain a better all-round ride with valves in the shock-absorbers which completely close the liquid passages until a given relatively low arm pull is reached, after which the shock-absorber functions in a conventional manner. This, however, is but a step toward the damping only of car-body oscillations of low magnitude and of high frequency, as illustrated by Fig. 7. In this chart the ordinate is in increments of length and the abscissa in increments of time. The curves represent the swing of an automobile body (DeSoto Six) taken over the rear axle and for an impulse of the same magnitude caused by lifting the axle 4 in. above the ground and allowing it to fall suddenly. The oscillations of the axle are not shown in this chart.

It will be seen that with the 5.25-18 tires, inflated to 35 lb. per sq. in., and conventional shock-absorbers, the body comes to rest after 1½ oscillations and no disturbance follows after this time period. With 7.50-15 tires, inflated to 18 lb., and conventional shock-absorbers, the slow swing of the body stops after one cycle, after which several minor fast oscillations are visible. These make themselves felt as an unpleasant jiggle in the car and are most noticeable at low speeds. With 7.50-15 tires, inflated to 18 lb., but with static-valve shock-absorbers, the body comes to rest after slightly more than one cycle and no further disturbance is noticed.

B. J. LEMON:—Mr. Hunt is interested in data that show variation of the rate of tires considered as springs and the hysteresis losses of deflected tires. The rate of change of load with deflection of the tire considered as a spring may be obtained from load-deflection curves and expressed as the change in deflection per 100-lb. change in load at the standard load of the tire. This value will express the increase in riding comfort obtained with each change to lower-pressure tires. I know of no values to show the hysteresis loss for the past and present tires.

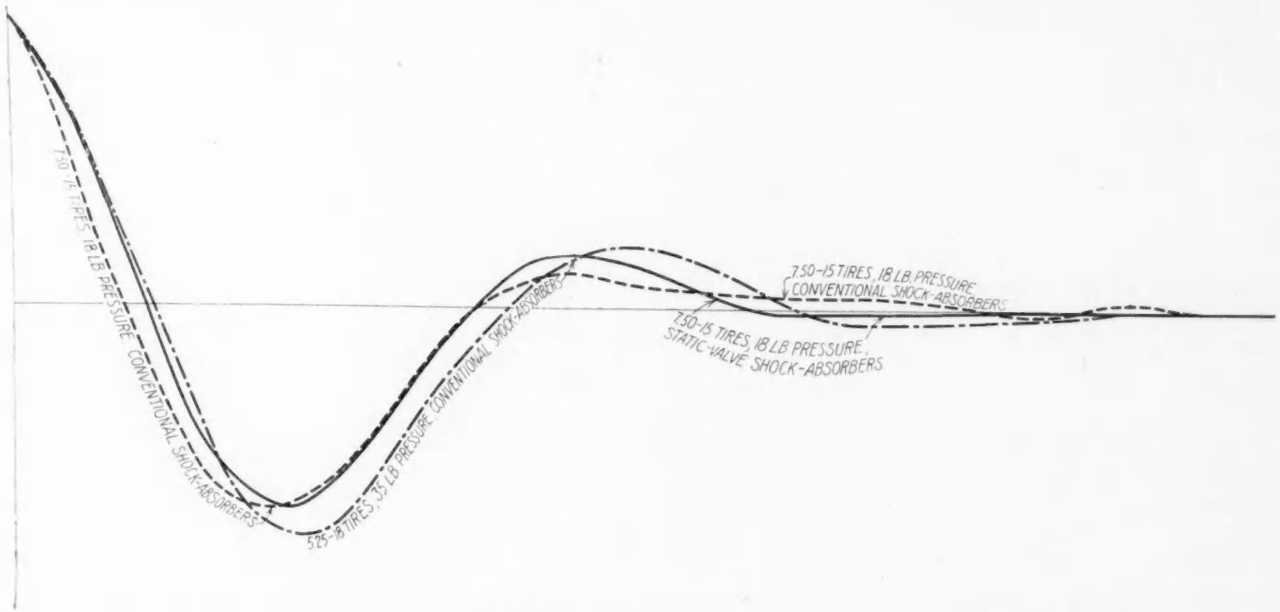

7.50-15 TIRES, 18 LB. PRESSURE, CONVENTIONAL SHOCK-ABSORBERS
7.50-15 TIRES, 18 LB. PRESSURE, STATIC-VALVE SHOCK-ABSORBERS
7.50-15 TIRES, 18 LB. PRESSURE, CONVENTIONAL SHOCK-ABSORBERS
5.25-18 TIRES, 35 LB. PRESSURE, CONVENTIONAL SHOCK-ABSORBERS

FIG. 7—CURVES OF CAR-BODY OSCILLATION WITH DIFFERENT TIRE AND SHOCK-ABSORBER COMBINATIONS
Least Body Disturbance Occurred with 7.50-15 Tires and Static-Valve Shock-Absorbers

Mr. Tuttle takes issue with my statement that shimmy and tramp tendencies are factors affecting all cars when equipped with super-balloon tires. We have found that several weeks' service was required to develop front-end instability in some cars. In this period, front-end looseness accentuated instability to a point that limited satisfactory car performance on some types of road. Perhaps there are cars that are invulnerable to such troubles over long service periods when equipped with super-balloons, but we have not discovered them as yet. My statement was as broad as my experience but not so broad as to include every make of 1932 car.

Relation of Design to Airplane Maintenance

Aeronautic Meeting Paper

By John G. Lee[1]

LOW-COST maintenance is secured by attacking the problem before the design is started. The author tells how this important feature can be designed into the airplane. Maintenance requirements should be written into the contract specifications which should indicate the time within which each part should be inspected and serviced. A suggested set of such specifications is submitted. By this procedure maintenance time can be cut in half.

The work of designing must not be rushed. To provide for quick maintenance, some broad changes from customary design are needed and will add to first cost but save money in the long run. Numerous recommendations are made as to design or type of important elements which will facilitate maintenance, avoid exasperation and add to passenger safety and comfort.

THE dominant factors in commercial-airplane design are speed, load, safety, passenger comfort, initial cost and cost of maintenance. Improvement in any one of these items almost invariably involves a sacrifice in one or more of the others. For example, the use of streamlines on the landing wheels increases the weight and the cost and adds to the maintenance but produces an increase in speed of 2 to 5 m.p.h.

Improvement is made in all of the basic factors over a long period of time, because of our ability to capitalize experience. Perhaps the designer's greatest weakness is that he does not make full use of the operating experience obtainable from previous designs, and for him to talk too much about how an airplane should be designed for easy maintenance may be unwise. He is likely to be asked, "Why don't you practise what you preach?" It is a fair question.

An airplane designer governs his design by the specifications given him. These always stipulate the required performance with great care and specify the equipment in detail, and the cost, of course, is guaranteed. Maintenance, however, is seldom mentioned. As a consequence, the designer does his best to exceed the specifications, so as to be on the safe side, and gives less attention to the items not mentioned. In the case of the Army and the Navy, the maintenance requirements are specified at considerable length in the designer's handbooks and other publications. This is not so with commercial lines.

Here, then, is the crux of the problem: the designer must be given a set of detail maintenance requirements. Like all good specifications, these must state what is desired but not how it should be accomplished.

In any maintenance requirement the time element enters vitally. For instance, to specify that the radio transmitter must be easily removable is not enough; one must specify that "the transmitter must be removable by one man in 4 min." Specifications of this type produce results. They will be complied with because the purchaser can demonstrate whether or not the airplane meets the specifications.

One of the bugbears of the designer is that maintenance men say little before an ariplane design is started but say plenty after the machine is completed. The maintenance department of every operating company should convince the executives of that company of the necessity of including specific maintenance requirements in every new specification.

An inexperienced executive may ruin the designer's efforts if he demands an abnormal rush to complete the design of a new airplane. Haste not only makes waste but means poor maintenance. All maintenance problems are essentially problems of detail design, and this cannot be rushed. Until recently, many operating executives lacked the familiarity with airplane handling that is essential to an appreciation of the complexity of an airplane. Two years is not too much to allot to the designing and putting in production of a new type. The fact must be remembered that good maintenance must be designed into the airplane; it cannot be applied afterward.

TABLE 1—OPERATING COSTS OF SINGLE-ENGINE AND TRI-MOTOR AIRPLANES

	Single Engine, Per Cent	Tri-Motor, Per Cent of Single-Engine Cost
Maintenance and Overhaul		
Airplane	25.9	36.9
Engine	15.3	34.5
Fuel and Oil	17.8	34.0
Insurance	8.4	13.8
Depreciation		
Airplane	19.8	36.9
Engine	12.8	24.0
Total	100.0	180.1

Units That Cause Major Maintenance Cost and Most Accidents

Formation of a clear idea of the relative importance of maintenance and the other factors in design is essential. Table 1 is an analysis of the direct operating cost, less pilot's pay, of one of the larger American airlines, which operates both single-engine and tri-motor equipment. Maintenance and overhaul of the plane, as distinguished from engine maintenance, represents about 26 per cent of the direct operating cost of single-engine airplanes. Depreciation of the plane represents an additional 20 per cent. The charge for depreciation is largely arbitrary, for it can be set high or low, depending upon the practice of the operating company. However, in the long run, depreciation is based upon maintenance. The direct operating cost of tri-motor planes is, in this instance, 80 per cent more than that of single-motor planes.

This may not present quite a true picture, because the basic factors of pay-load and cruising speed, and the more intangible factors of passenger comfort and safe-

[1] M.S.A.E.—Chance Vought Corp., Hartford, Conn.; formerly project engineer, American Airplane & Engine Corp., Farmingdale, N. Y.

ty, do not appear. These are contributions to revenue and not to cost. Moreover, we have no way of evaluating either comfort or safety; although safety is dependent chiefly upon good design and good maintenance, it cannot be expressed in terms of dollars and cents.

Maintenance costs of another airline have been broken down in Table 2 and apportioned to the different units in the airplane. That items such as wings and fuselage should carry so large a percentage of the cost is perhaps surprising, but the fact must be remembered that the expense of these parts comes at the time of overhaul, not in line maintenance. The expense is incurred, even though one seldom sees a mechanic at an airport working on the wings.

Another interesting viewpoint is obtained from Table 3, in which the causes of airline accidents that can be attributed to the airplane have been itemized. Two facts stand out: First, the powerplant is responsible for 50 per cent of the accidents and the landing-gear units for 35 per cent more. I think that powerplant accidents are largely dependent upon maintenance, which reflects on the designer, who usually designs a crowded installation. Landing-gear accidents are attributable directly to the designer. Department of Commerce requirements for landing-gears apparently are not severe enough, and we would be wise to design our landing-gears to absorb more severe shocks than are specified by the Department.

Ways to Reduce Wear in Moving Parts

Operating troubles may be classified under the general headings of wear, deterioration and fatigue. Wear can be reduced in four ways. The first of these is by the use of ball bearings on all moving parts such as control-surface hinges, control-system levers, pulleys, engine controls and the like. These bearings should be of the cartridge type, packed with grease at the factory and sealed. Their use is not as expensive as is generally supposed, because they prevent looseness in the joints, which often require hand fitting and reworking on final assembly. Second, oilless bushings on parts such as landing-gear fittings are recommended. Several types are on the market. If an oilless bushing cannot be used, a standard grease-gun fitting is necessary. Bushings should be designed to turn on the bolt, so that both wearing parts can be replaced. This is not the usual practice, because bushings frequently are used as spacers between two thin bearing surfaces which are clamped tightly to the bushing with a through bolt. Both bushings and ball bearings should be readily replaceable.

Close clearances in all moving joints will reduce hammering in them and greatly prolong their life. Clevis bolts are frequently used because the manufacturing tolerances are closer than with standard bolts. One of the chief sins of the designer is to calculate the bearing strength of his material at its ultimate value; that is, he figures only sufficient bearing to prevent ultimate failure at full design-load. He should design the bearing loads on the yield point of the material in tension to correspond to the maximum imposed load, including load factor. This will assure that the working loads are only about 20 per cent of the yield point and will greatly prolong the life of all parts. The penalty in increased weight is not as large as is generally supposed.

TABLE 2—APPORTIONMENT OF MAINTENANCE COSTS BASED ON 28,000 FLYING HR. ON THREE TYPES OF MULTI-ENGINE AIRPLANE

	Per Cent
Wing	16.0
Fuselage	20.3
Cabin	15.2
Control System	8.2
Landing-Gear	13.2
Instruments	5.0
Fuel System	7.0
Oil System	1.2
Engine Controls	2.6
Cleaning	1.9
Miscellaneous	9.4
Total	100.0

TABLE 3—SOURCES OF ACCIDENTS ATTRIBUTABLE TO AIRPLANE[a]

Powerplant	Per Cent	Structure	Per Cent
Fuel System	9.8	Flight Controls	3.9
Cooling System	0	Movable Surfaces	0
Oil System	3.9	Stabilizer and Fin	3.9
Ignition System	2.0	Wings	3.9
Engine	3.9	Landing-Gear	31.4
Propellers and Accessories	7.7	Wheels, Tires and Brakes	4.1
Miscellaneous and Undetermined	21.6	Fuselage and Engine Mounts	3.9
		Miscellaneous and Undetermined	0
Total Powerplant	48.9	Total Structure	51.1

[a] Based on Department of Commerce Air Commerce Bulletin, vol. 3, no. 17, covering 65 accidents, July to December, 1931. Accidents attributable to airplane constituted 40 per cent of total accidents.

Applying the Painters' Slogan

Deterioration can be delayed by correct choice of materials and by protecting those materials which otherwise will not withstand weather or abrasion. The painters' slogan, "Save the surface and you save all," applies in this case, whether the surface is protected by cadmium plating, case-hardening varnish or dope. Finishes are much heavier than is generally realized. A good dope and paint job, such as is desirable for ease of cleaning as well as for appearance, may weigh 90 lb. for a 10-place airplane for the dope and paint alone. The Army quotes 0.070 lb. per sq. ft. for doped fabric; for commercial work, 0.096 lb. is not unusual.

Certain standard finishes, such as cadmium plating, should not be used on fittings to which washers are welded as a reinforcement, as the electrolyte from the plating gets down between the two pieces and is almost impossible to remove. The plating acts in this case as an accelerating agent to hasten corrosion instead of to prevent it. These built-up welded fittings present a real problem. Apparently the best solution is to Parkerize them, then dip in slushing oil and bake on enamel. Even this leaves something to be desired between the adjacent surfaces.

Perhaps the best way of dodging the protection procedure is to use non-corrosive materials, such as bronze or stainless steel, throughout. These are intrinsically more expensive, but in many instances the cost of the material is offset by the saving in the protection process. Probably we shall in the future make much more use of the free-machining stainless steels for fittings and other machined parts.

The question of protection for duralumin calls for a whole paper by itself. It will suffice to state herein that the anodic treatment or the use of Alclad, together with appropriate paint protection, is about the best we can do. In all cases, however, the use of grease on duralumin as well as on steel does considerable to prevent corrosion. Any protective material used should have some pigment in it so that a man can tell by the color whether the protective coating is intact or not.

Field Experience Needed to Avoid Fatigue Failures

Fatigue of metal presents a very intangible difficulty. No one can foretell when a crack will start or detect it until it has progressed for some distance. The designer can be guided only by experiences in the field as to those things which most frequently cause fatigue failures. Here, especially, close cooperation between the maintenance men in the field and the designer in the factory is required, as even the best design can be rendered dan-

gerous if proper experience is not brought to bear in the installation of parts. A few examples of fatigue difficulties may refresh our minds.

Failures of fuel and oil lines as the result of fatigue are rather common. Every fuel or oil line must be flexibly mounted so that it can yield freely and not transmit vibration; yet it cannot be too flexible or its own weight will set up a vibration that is quite as serious as any transmitted from the engine. Annealed copper or aluminum tubing is usually employed. This should be re-annealed after bending and at intervals thereafter, to prevent age-hardening. Recent attempts to eliminate this objectionable requirement have led to the use of flexible metal tubing and of fatigue-resisting bronze tubing.

A typical case of a mounting that must be flexible but not too flexible is that of the instrument-board. On cabin airplanes the instrument-board is very close to the engine, and the instruments will not withstand the vibration if the board is rigidly mounted. On the other hand, the board and instruments may weigh 10 to 15 lb., which is a sufficiently heavy mass to acquire a vigorous vibration of its own. Rubber mounting is likely to be too free; felt has been found to afford an excellent compromise.

Fatigue manifests itself also in the fraying of control cables wherein individual strands are broken as a result of reversal of stress in repeated bending over small pulleys. Preformed cable represents a distinct advance. Fatigue failures sometimes occur in streamline tie-rods and in certain instances may be very serious. A single landing with a broken fin-wire will convince anyone of the danger. I highly recommend designing the tail or any other wire-braced structure so that breakage of any one wire will not imperil the machine.

In providing for streamline wires, the terminals should be of the universal type, which affords the only means of securing perfect alignment of the wire. If the wire terminals can swivel in only one direction, as is the case with the present Army-Navy standard, there is no way of making sure that sooner or later the terminal fittings may not be bent out of line with the wire, setting the stage for a fatigue failure. Even the brass clips that are slipped over streamline wires as identification tags will localize fatigue, and, together with corrosion between the tag and the wire, may cause a failure. The Army has now eliminated these tags.

Concentration of stress caused by an abrupt change in the section of a machined part has long been known to invite fatigue failure. This is particularly true of threads, eye-bolts and similar parts. It is in these details that some important point is often overlooked in hasty design.

Simplicity Facilitates Maintenance

One of the easiest means of simplifying maintenance is to omit unnecessary units. This is the only way in which weight, first cost and ease and cost of maintenance can all be improved at once. An excellent example is the full-swiveling tail wheel. Some designers take great pains to provide a spring mechanism to keep the tail wheel aligned in the direction of flight. This mechanism makes the handling of the plane on the ground so

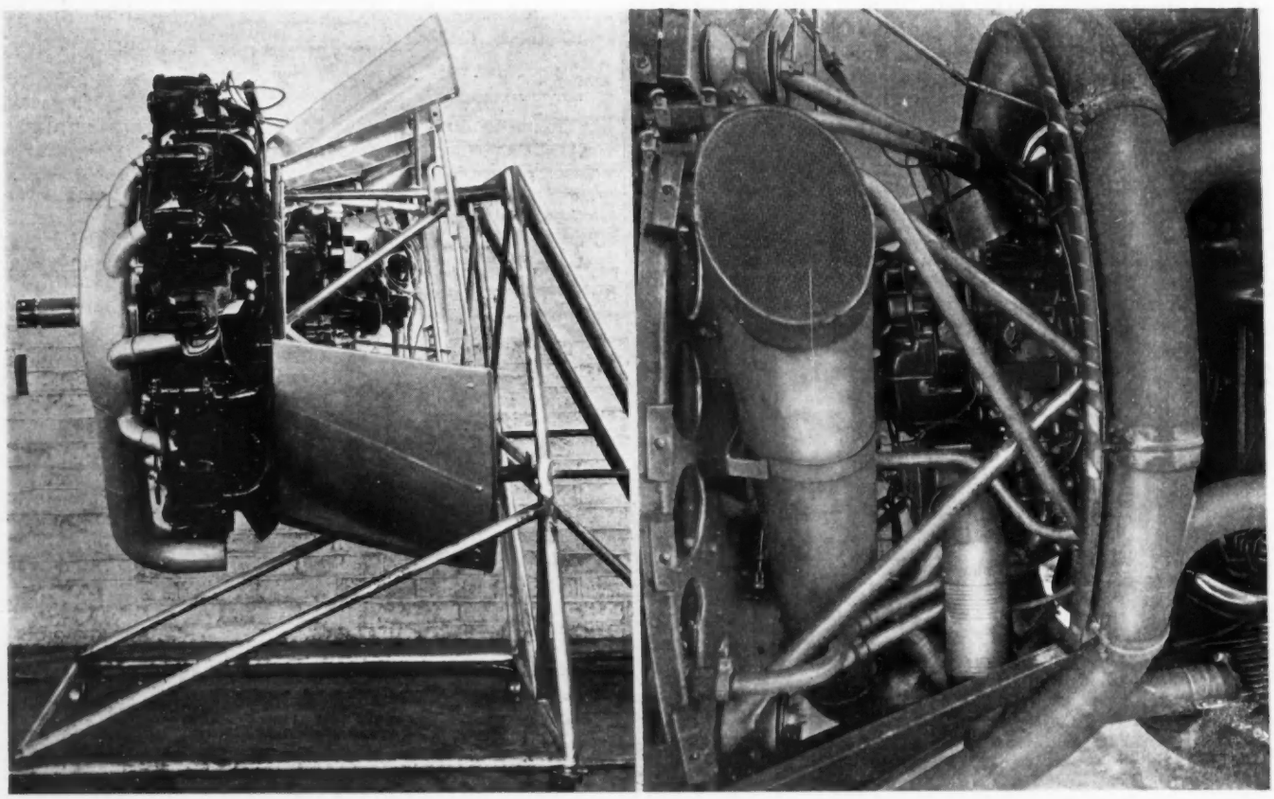


FIG. 1—DETACHABLE POWERPLANT ON WORK STAND AND IN AIRPLANE

(Left) A Wright Cyclone Engine on a Pilgrim 100-B Airplane Mounting Showing How Clean Installation Is Made Possible by the Front Exhaust Manifold and the Single Cowling with Buttons Only at the Rear. (Right) A Pratt & Whitney Hornet on a Pilgrim 100-A Airplane. In This View the Convenience of Detaching the Whole Assembly as a Unit Is Evident. Rubber-Insulated Mounting Cones Are Shown at the Extreme Left. Note Rear Exhaust-Ring and Carbureter Riser

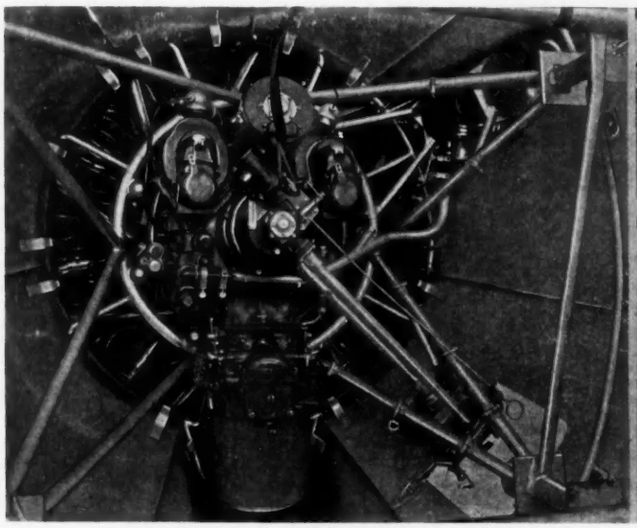


FIG. 2—REAR OF POWERPLANT, SHOWING ACCESSORIES

Demounting the Accessories toward the Rear Is Very Difficult When the Engine Is in the Fuselage. Note Cowled-in Oil Cooler at Top Right, External Inertia Starter at Bottom Right and Spring Cowling-Clips

FIG. 3—EXHAUST MANIFOLD AND CABIN HEATER

Cold Air Is Taken from in Front of Engine Cylinders and Passes through a Seamless Stainless-Steel Pipe inside the Exhaust Bayonette, at the End of Which Is a Heater Control-Valve

difficult that release cams have been devised to free the tail wheel from restraint. All of this complication can be eliminated if the wheel is allowed to swivel about an axis that slopes forward about 3 deg. at the top when the tail wheel first makes contact with the ground. Thus, the weight of the airplane keeps the wheel in line when taxying, since swiveling to either side automatically raises the tail slightly, and friction is sufficient to keep the wheel aligned in the air.

Purchased parts and common accessories give much trouble to both designers and maintenance men. Door locks especially are a source of annoyance. No really satisfactory locks that are light, compact and serviceable seem to be available. Parts and materials should always be purchased from reliable companies that can be expected to remain in business, as nothing is more of a nuisance than trying to obtain special parts from defunct companies, unless perhaps it is the "lame duck" airplane. A "lame duck" airplane is one that differs from all others of its type and the parts of which are interchangeable in name only. Usually it is the first of its series and is the inevitable by-product of haste.

Proposed Specification of Maintenance Requirements

I have stated that no detailed maintenance requirements are contained in the specifications for most commercial designs and also that such maintenance requirements must specify the time in which a given piece of work can be accomplished. With the intention of provoking discussion on this subject, I have prepared the following tentative set of specifications. I do not pretend that these specifications are exhaustive, and they may even show signs of a designer's ignorance of maintenance problems. However, they do contain the time element and will at least serve as a good working sample. Criticisms and additions are in order. The figures in parentheses give the number of men and the time required to execute the operation.

I—POWERPLANT

(1) *Entire powerplant*, including oil system, cowling and manifold, to be quick-replaceable (3 men, 30 min.)
(2) *Exhaust manifold* to be interchangeable in sections. Stainless steel, non-cracking. Any section replaceable without demounting other sections (1 man, 10 min.)
(3) *Oil-system* parts must be supported to absorb vibration. Oil coolers must be above and drain into oil tank. Filler cap and drain plug must be tight and quick-removable (1 man, 30 sec.)
(4) *Fuel-system* lines must all be removable without taking airplane to pieces. Any line to be removed completely (1 man, 15 min.). Combined fuel-system unit removable (1 man, 30 min.). Each fuel tank independently removable (2 men, 45 min.). Cocks on all tank outlets. All connections readily inspectable (1 man, 3 min.). All tank-filler necks large and accessible. Tanks to be filled at 20 gal. per min. without spillage. Fuel strainer removable (1 man, 45 sec.)
(5) *Engine controls* to be push-rod and ball bearing throughout, including shutter, carbureter heater, etc. Each control system to contain one lever of adjustable length. All rod lengths adjustable and stiff enough to take 70 lb. on the control handle without failure. Adjustable stops should be provided
(6) *Engine cowling* complete, including anti-drag ring, to be strictly interchangeable and quick-removable (1 man, 4 min.). Every piece to be independently removable without affecting adjacent piece

II—LANDING-GEAR

(1) *Wheel cowling*, if present, to be taken off in one unit (1 man, 5 min. per wheel)
(2) *Shock-absorber* struts to be inspectable for leaks and fluid quantity without removing cowl (1 man, 6 min. per airplane)
(3) *Bearings* to be replaceable and self-oiled
(4) *Tail wheel* to be full-swiveling (360 deg.) and inspectable from side or rear, not from beneath fuselage. Fuselage to be divided from tail-wheel compartment by water-and-dustproof wall
(5) *Towing links and jacking pads* to be provided at tail and each wheel outside all cowling. Jack pads to be located so that jack can be used when tire is flat, especially with air wheels
(6) *Retractable landing-gears* must be completely inspectable in down position and capable of being locked in down position to permit operation of airplane with retracting mechanism inoperative or absent. Jack pads must be provided for use when jacking up entire airplane and permit operating landing-gear (2 men, 30 min.)
(7) *Brakes* must be operable by mechanic while pushing the airplane. Parking brake required. Brake shoes to be adjustable without removing cowling

FIG. 4—A SIMPLE TAIL-WHEEL ASSEMBLY

The Fork Can Swivel Freely through 360 Deg. in a Frame Pivoted to the Fuselage. The Shock-Absorber Oleo Attaches to This Frame. Shimmy Is Prevented by Inclining the Swivel Axis Forward 3 Deg. from the Vertical

III—
PILOT'S COCKPIT

(1) *Instrument - Board* must be fully accessible from the rear (1 man, 2 min.) and removable as a unit (1 man, 25 min.)
(2) *Pilot's controls* must all be accessible and provided with adjustable stops. Each control (elevator, rudder, brake, throttle, etc.) must be an interchangeable unit and quickly replaceable (1 man, 10 min. each)
(3) *Piping and push-rods* to be removable without dismantling other parts
(4) *Complete cockpit inspection* to be made possible (1 man, 10 min.)

IV—
PASSENGER CABIN

(1) *Cabin seats* to be all identical and quick-removable (1 man, 2 min. per seat)
(2) *Interior finish* to be washable in place. Upholstery panels to be removable for fuselage inspection (3 men, 4 hr.). Inspection doors to be provided for all control lines and piping but not operable for passengers (complete inspection, 1 man, 10 min.)
(3) *Heating and ventilating ducts* to be removable without tearing down airplane. All joints and passenger fittings to be accessible for repairs
(4) *Wash room* to be washable with hot water and provided with drain holes. Complete servicing, including fresh drinking and washing water (1 man, 5 min.)
(5) *Rear fuselage* interior accessible through door which is non-operable by passengers. Catwalk to be provided in certain cases
(6) *Windows* shall be non-opening and quick-replaceable (1 man, 30 min. per window)
(7) *Baggage, mail and fuel* shall all be loaded without entering passenger cabin. Separate doors shall be provided for pilot's cockpit

V—CONTROL SYSTEM

(1) *Bearings* shall be replaceable, either ball or roller, packed in grease and factory sealed. This applies to all levers, pulleys, hinges, etc. Take-up shall be provided on control-column bearings
(2) *Adjustable stops* shall be provided at the pilot's cockpit and at the control surfaces
(3) *Inspection doors* shall be large and accessible for all pulleys, terminals, etc. All control-system doors must be capable of being opened and complete inspection made (1 man, 8 min.)
(4) *Stabilizer adjustment* must be of irreversible-screw type, fully enclosed, including the threads. Torque-tube bearings must be self-lubricated, non-binding and non-rattling.

VI—ELECTRIC SYSTEM

(1) *Electric wires* shall be run in metal conduits to junction boxes and suitably marked at both ends to correspond to the wiring diagram. All wires shall be removable and replaceable by pulling through conduit
(2) *Battery* shall be mounted so as to be accessible from outside the airplane. Must be capable of being watered and tested in place (1 man, 3 min.) or replaced complete (1 man, 4 min.)
(3) *Radio* units shall all be accessible and removable from outside of airplane without entering the cabin. Transmitters, receivers and dynamotors must each be replaceable (1 man, 3 min.)
(4) *Landing lights* must be accessible at rear side. Bulbs must be replaceable (1 man, 2 min. each)
(5) *Fuses* must be replaceable in flight with one hand

VII—WINGS

(1) *Glued joints* will not be permitted
(2) *Metal-covered wings* shall be provided with removable sections to permit complete inspection of interior without reconstructing wings
(3) *Provision for adjustment* shall be made to accommodate wing heaviness. Complete adjustment (2 men, 20 min.)
(4) *Attachment fittings* shall be designed so that close fore-and-aft fit is required on one spar only, with clearance on other spar. All joints shall be readily inspected (1 man, 5 min. per airplane)

VIII—TAIL

(1) *Hinge pins* must be accessible. Elevator and rudder must be quick-replaceable (1 man, 10 min. per surface)
(2) *Stabilizer* should be rigidly attached and strong enough to use for pushing airplane
(3) *Fin adjustment* to be made without disconnecting fin braces and shall be of the screw type. Fin readjustment (1 man, 5 min.)
(4) *Lifting means* shall be provided to accommodate from 6 to 10 men. Socket for lifting bar preferred rather than handles
(5) *Tail cowling* to be quick-detachable (1 man, 30 sec. to uncover tail wheel, stabilizer adjustment and control-surface horns)

IX—GENERAL

(1) *Finish* of dope and paint to be smooth, for easy cleaning. Non-fading colors to be used which can be matched. No plating to be used where electrolyte can seep into

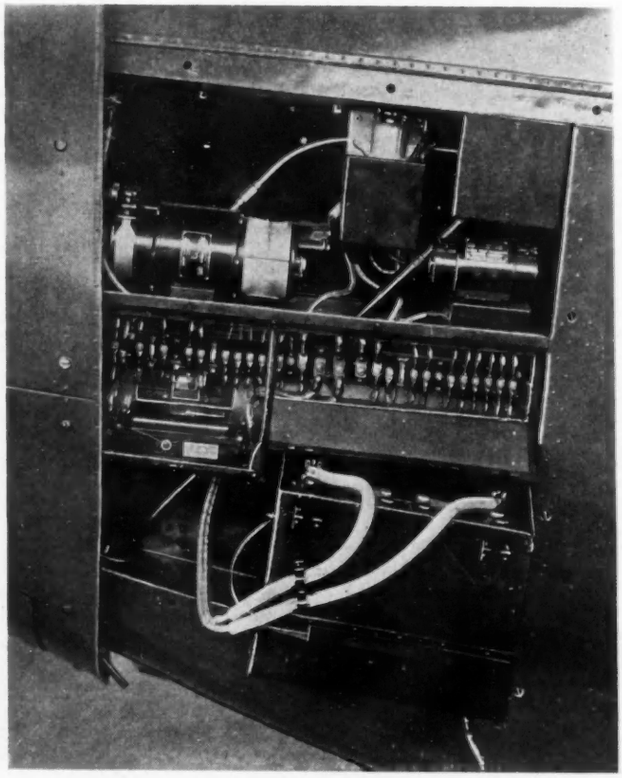


FIG. 5—EASY-SERVICE ELECTRIC-UNIT COMPARTMENT

Showing Radio Dynamotors, Main-Terminal Panel, Generator-Control Box and Battery. The Battery Is Supported on a Sliding Track

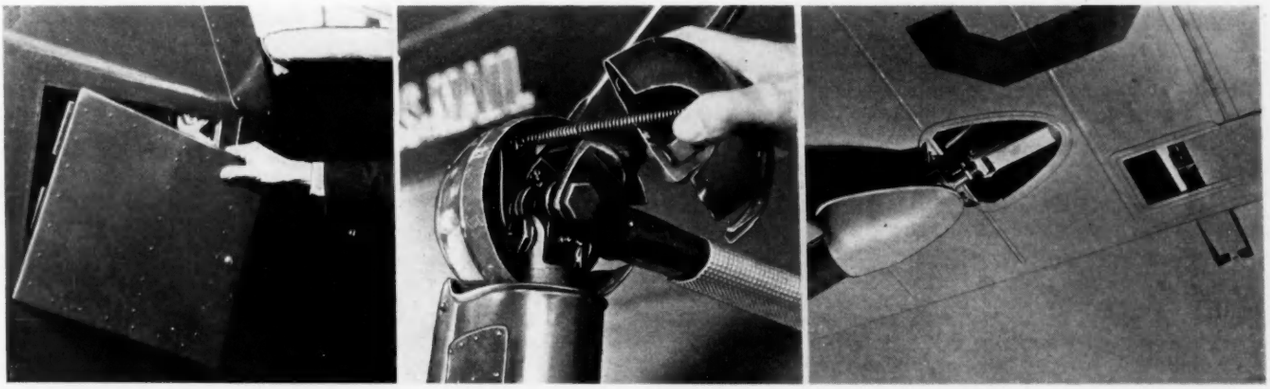


FIG. 6—SOME DETAILS THAT FACILITATE INSPECTION AND SERVICING

(Left) Fuselage Inspection Door Having Only One Cowl Fastener and Catches at Front That Fit Inside Fuselage Frame, Locating the Door and Holding It Snug. (Center) Fairing at a Joint. The Front Cap of the Streamline Cover Is Held in Place by a Coil Spring and Is Removable without Tools To Expose the Fitting, Which Has Grease-Gun Connections. (Right) Inspection Doors on a Wing. The Metal Track Is Riveted to a Celluloid Frame and the Whole Assembly Is Doped onto the Cloth

interior of parts or plating may be worn off. Drainage shall be provided to remove moisture from all pockets

(2) *Steps, doors and reinforcements* shall all be strong enough to withstand abuse and conveniently located in the opinion of the customer

(3) *Instructions and diagrams* shall be contained in a complete instruction manual giving part numbers of all principal assemblies and all detail wearing parts. Diagrams covering rigging, fuel system, oil system, electric system, radio installation and loading instructions should be furnished in this manual and posted in the airplane at convenient points

(4) *Hulls and metal monocoque* fuselages shall be accessible at all interior points for inspection and riveting. No closed members or blind rivets shall be used. All interior structure shall be capable of being laid bare (3 men, 4 hr.)

No airplane meets these specifications at present and none ever will meet them unless the maintenance specifications are put on an equal footing with the performance specifications. The requirements are not impossible to meet, but no designer will meet them if he is not required to do so. And when he does meet them, certain sacrifices, principally in first cost, will be involved.

Influence of Maintenance on Design Features

When maintenance requirements are designed into the airplane in the beginning, they exert a profound influence on the design. Perhaps the first consideration is the space available between the engine-mounting ring and the front wing-spar. This is the most valuable space in the airplane. In it are the engine installation, pilot's cockpit, surface controls, instruments and frequently cargo space. Every effort should be made to keep the distance from the engine ring to front spar as large as possible, even to the extent of lengthening the tail to keep the airplane balanced. Crowding this space complicates the designer's problems, increases the shop cost and is ruinous to maintenance.

The quickly detachable engine installation is another requirement that has a widespread effect. Obviously, the oil tank and the oil cooler must be part of the detachable mount. Likewise the cowling must be anchored to the mount and not to the fuselage. The engine controls must be specially designed so as not to require the undoing of a large number of bolts to get the rods apart. All electrical wiring must be brought to a master plug having multiple contacts that can instantly be plugged into the mating receptacle on the fire wall. These things are not hard to do if they are included in the original concept of the design but are impossible to do afterward.

I cannot too heartily endorse the advantages of the detachable engine mount, which are self-evident on the wing engine of a multi-engine airplane. When the engine attaches to the fuselage, the advantage is even greater. The detachable mount means that a group of workmen building the airplane can get at the fire wall from in front, while another group works on the engine from behind. The engine may be mounted on a stand at a convenient height from the floor instead of being bolted to the airplane at a height requiring stepladders. All of this results in reduction of shop costs.

At first sight it would seem that the jigs needed to make the engine mounts interchangeable on different fuselages would offset the gain. This is not so. In the Pilgrim airplanes the engine mount is attached to the fuselage through the medium of hollow rubber cones. The clearance between attaching bolts and their bolt holes is about $\frac{1}{8}$ in. on the radius. No metallic contact occurs, no accurate jigging is required and no unyielding vibration is transmitted.

Engine Parts and Accessories Inaccessible

Engine builders clearly are at fault in the arrangement of the rear of their engines. Virtually all accessories are arranged to be removed toward the rear. This applies to generators, starters, magnetos and so on. When the engine is installed in the airplane, only a contortionist can remove the generator nuts and pull out the generator. These units should be designed to be removed from the side.

Another feature of engine design which complicates maintenance is the combination of the exhaust manifold and carbureter intake-heater. On some engines these units actually obstruct one-half of the working space around the engine mount. No doubt the newer type of injection engine will dispense with some of this complexity. The location of gasoline and oil strainers in most radial engines also is very inconvenient.

The exhaust manifold is perhaps the most annoying part of the engine installation. Every engine designer is tempted to make a streamline manifold of some sort, but sooner or later these crack at the weld as a result of unequal expansion. If the designer does not provide a manifold of circular section with ample expansion joints, the maintenance men do it for him later. When designing a manifold, remember that expansion means not only

a change in length of the pipe but also a change in angle. Therefore all joints must permit a certain degree of articulation. While stainless steel prevents the burning out of manifolds, it increases all of the other difficulties because of its high expansion coefficient and its resistance to welding.

The Cabin-Heater Problem

Closely associated with the question of the exhaust manifold is that of the cabin heater. The conventional solution is to put an air jacket on the manifold and conduct the heated air to the cabin. The usual result is an objectionable odor in the cabin. A tendency of late has been to use steam boilers or Prestone heaters, which are piped to the cabin, where some form of radiator is used. While this is a promising field, the system has not yet been developed sufficiently as regards convenient maintenance and the weight is also excessive.

Perhaps the best solution at present is the hot-air heater, which obtains fresh air from just back of the propeller and forward of all sources of oil or odor from the engine. This air is then carried through a pipe enclosed in the exhaust pipe and thence to the cabin heater. This works very well on the Pilgrim airplanes. It has two advantages; first, more heat is obtainable in cold weather, as the air pipe is surrounded by hot exhaust gas, whereas the conventional air jacket is surrounded by cold air; and, second, the air pipe is small enough to be made seamless, which means that there are no welds to crack and no possibility of carbon-monoxide gas entering the cabin and causing airsickness or poisoning. In the Pilgrim installation this internal pipe is clamped into the manifold to permit relative expansion of the hot manifold and the cool pipe. Provision is also made to bypass the hot air to the atmosphere when it is not required. Thus a circulation of cool air is always maintained through the exhaust pipe and there is no danger of its burning out.

Design Treatment of Rear-End Units

Turning now to the rear end of the airplane, we have another group of units which require maintenance and are usually awkward to reach. These are the tail wheel, the stabilizer adjustment and the control-surface horns. Probably the most satisfactory way to treat these is to terminate the fuselage at a wide bulkhead instead of a narrow tail-post. The rudder should be wholly above this bulkhead and the elevators entirely to the sides. The tail-wheel unit and the stabilizer-adjustment housing should bolt directly on the face of the bulkhead. The entire assembly would be covered by a fairing cap, removable upon releasing a single catch. This cap would also enclose the control-surface horns. The Lockheed monoplane and some others have approached this type of design.

Certainly the most difficult problem with which the designer is now faced is to reconcile good maintenance with a monocoque fuselage, which depends for its strength upon its unbroken shell surface. The openings required for windows, entrance doors and inspection doors seriously impair its strength and call for complicated construction. But, since the monocoque has such outstanding advantages in speed and headroom, we shall be forced to accept it sooner or later. Only two practical compromises seem to be possible. One is to end the fuselage shell just back of the pilot's compartment and treat the front end as a welded tubular structure with detachable cowling. The other is to build an inner and an outer shell at the front end, the former to carry the load and the other forming a removable fairing. Space between these two shells would be available for electric installations, oil tanks and so on. Personally, I favor stopping the fuselage shell back of the pilot's cockpit.

The construction of a monocoque fuselage presents a design problem of the first class. A safe assertion is that at present no part of a monocoque construction can be replaced; it can only be rebuilt. Too little experience has been had to date to draw conclusions regarding the best type of construction, but that some of our worst future maintenance problems will center about this point is certain.

Retractable Landing-Gear Maintenance

One more problem that is now being fought out is that of the high-speed landing-gear. The retractable gear is the best as regards speed, but no absolutely reliable retracting mechanism has really been designed, although some promising experimental types are under way. The maintenance problems involved with a retractable gear are severe. The whole situation may be regarded as still strictly experimental.

On the other hand, the cantilever gear installed on the recent Curtiss pursuit planes and on the General Aristocrat of a few years ago is much better from the maintenance standpoint. Gears of this type can be made almost as light as a standard gear. The stress and bearing loads, however, are very high and call for careful design. Streamlining the wheels is of much greater advantage on a cantilever gear than on the conventional type because of the lack of air-flow interference from a multiplicity of tubes. However, the streamline cover is difficult to maintain, not only because it vibrates in the air and on the ground, but also because it collects mud and ice. A successful streamline housing must be quickly removable as a unit and very solidly supported.

Maintenance Labor Can Be Halved

No one man can know all about design, operation and maintenance. The designer is only human in this respect and must depend upon second-hand information from the experience of the operating personnel. Therefore he must be provided with definite maintenance requirements in the specifications for a new design. It is imperative that these requirements should be a step in advance and not a duplication of present maintenance procedure. It is likewise imperative that they be so drawn that it can be demonstrated whether or not the air-airplane fulfils them. A criticism of the maintenance features of an airplane reflects as much upon the operator who ordered the airplane as it does upon the company that designed and built it.

No reason exists why airline maintenance and inspection cannot be reduced to one-half the present labor requirement. Neither is there any reason why the overhaul cannot be forestalled until curiosity, not deterioration, is the governing factor. To accomplish this we must put maintenance requirements on a par with performance requirements.

Finally, the company that designs an airplane must be given a reasonable time in which to execute the requirements of the airline, and the airline must be willing to absorb in increased first-cost the sacrifice that good maintenance demands.

THE DISCUSSION

CHAIRMAN R. C. MARSHALL[2]:—We have several times looked over the product that Mr. Lee has turned out, and I think that, from the maintenance standpoint, it is one of the best designed airplanes that have come out recently which I have examined. It shows the results of long, hard study and work and represents an effort to lower the cost of transport operation.

[2] President, Trans-American Airlines, Detroit.

GEORGE A. PAGE, JR.[3]:—I want to compliment Mr. Lee on his extremely good paper and what I closely observed as the excellent method of carrying out his own principles. Having had occasion to work on similar problems myself, I can appreciate the work that must have been behind this.

EDWARD P. WARNER[4]:—I would like to join in the compliments to Mr. Lee and the organization he has represented here for the work they have done, especially on the fundamental idea, which the paper embodies, of preparing maintenance specifications. That is an idea in which considerable interest has been taken and to which more public attention has been given in the last few months, I think, than ever before, although it is an old story with the military services and is gradually becoming a familiar one to transport lines.

A few months ago we published in *Aviation* a series of several articles by Ralph Lockwood, chief engineer of Eastern Air Transport, on just such a set of specifications for maintenance from the point of view of the operator. Mr. Lockwood listed about 100 operations and gave the number of men and minutes he thought should be required for each operation. One fact that struck me rather forcefully is that the same operations have not been selected by Mr. Lee. My impression is that in only a surprisingly small proportion of the large number of operations listed has the operator's selection for specification been duplicated by Mr. Lockwood and Mr. Lee. That seems to me rather sinister, because it suggests the infinite complexity of the subject and the difficulty of arriving at any definite conclusions.

Suggests Aeronautical Chamber Aid in Work

We have long since agreed upon the specification of most performance qualities and now are approaching the point where we can agree upon the definition of cruising speed and how to measure it, but we are very far from having any standard definition of maintenance quality. A long step in advance is taken when one person prepares a set of maintenance specifications. When two or three persons prepare them from different points of view, we have another long step to take in reconciling them. That suggests to me that this is a work which could be done effectively by the Air Transport Division of the Aeronautical Chamber of Commerce of America, as it is an activity that calls for cooperation among the operators, with at least the passive attention of the manufacturers during the time it is being inaugurated.

It is not too much to hope that we may lay down a general specification of maintenance qualities, at least for a broad class of planes, say for single-engine transport planes of five-passenger capacity or more, so that we may get a sort of coefficient of maintenance by combining many of these operations. As Mr. Lee has suggested, we often have to sacrifice weight or performance to get maintenance qualities. Similarly, it may be necessary to sacrifice one maintenance quality to get another, and all who study airplanes in detail will always

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[4] M.S.A.E.—Editor, *Aviation*, New York City.

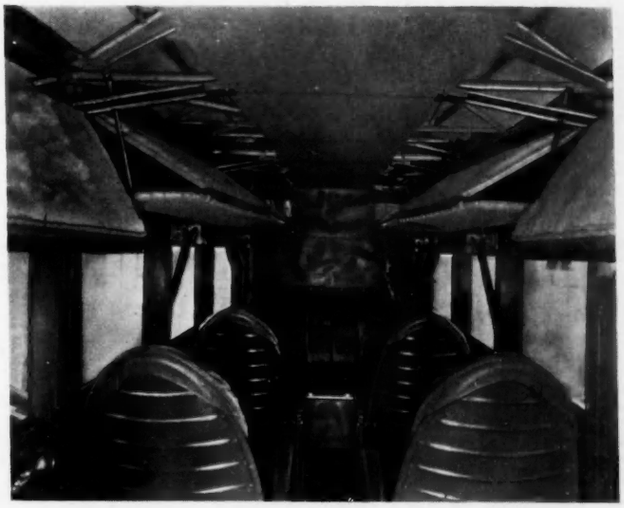

FIG. 8—INTERIOR OF CABIN, SHOWING OPEN INSPECTION PANELS IN CEILING

Leveling Lugs Are Clamped to the Structure and Are Reached through These Panels

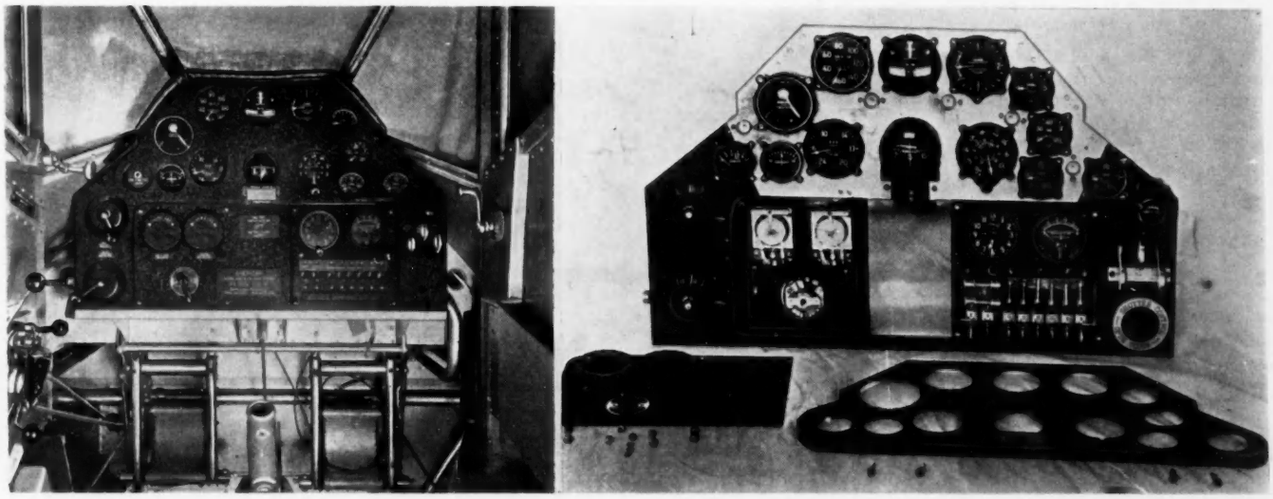

FIG. 7—INSTRUMENT-BOARD IN A CABIN PLANE AND TAKEN APART

The Whole Board Is Supported at the Bottom by a Felt-Lined Channel. Flying and Engine Instruments Are in a Lighted Panel at the Top; Radio Controls at the Bottom Left. A Hinged Switch Panel Is at the Bottom Right of the Panel. In the Right View the Guard Shield Is Removed from the Lighted Instruments. Mounting Screws Are Tapped into the Board So That Instruments Can Be Removed without Reaching Back of the Board. The Hinged Switch Panel Has Been Swung Forward To Expose the Fuses

give attention to the detailed requirements of an individual operating company.

Nevertheless, I think it would be worth considerable to have some understanding among the operators regarding the relative importance of these various qualities and to develop a sort of score-card based on points that can be added up to give a figure representing the over-all maintenance quality of any particular airplane. We shall have to standardize the mechanics, perhaps, as well as the requirements, if specifications are to be written in man-hours.

Buy New Planes to Reduce Cost

Another thing that this paper suggests is that the time has come to buy new airplanes. Table 1 of the paper shows that depreciation is 32.6 per cent of the direct operating cost for the single-engine plane and about 34 per cent of the total for the tri-motor plane, or virtually one-third of the total cost in each case. The total maintenance cost is virtually 40 per cent of the whole operating cost in each case; that is, maintenance is about one-quarter more than all depreciation, and fuel and oil is a little more than one-half as much as depreciation.

Airplanes have been flown for varying lengths of time and varying depreciation standards have been adopted. Let us assume two hypothetical cases, both of which are well within the range of possible experience. Suppose (*a*) that an airplane be depreciated on a 3-year basis, and (*b*) that it be depreciated on a 4½-year basis, In the first case the depreciation charge is increased relatively by one-half of the smaller of the two amounts. If the maintenance cost could be reduced by 35 per cent by putting in new airplanes, we should be well justified in discarding three-year-old planes that might give good service for another two years, deliberately shortening the service life by two years, and buying new machines. This takes no account of the possible saving in fuel cost by virtue of the greater economy and efficiency that should accrue to airplanes of modern design.

Regarding the specification of maintenance in the light of operating experience, I wish that Mr. Lee would tell us the approximate size of the single-engine plane to which the figures in Table 1 apply and whether it is an open mail plane or a cabin type. Also I should like to have him comment upon the relative standards of military and commercial maintenance. Some suggestion has been made that the air-transport companies still have much to learn from the Army and the Navy about maintenance standards. I think that the Army first became much excited about maintenance eight or nine years ago and began to prepare such specifications and to be very insistent upon them. As a result, the maintenance quality of the Army airplanes improved rapidly through a period of three or four years, when perhaps it received more attention than the inherent performance of the plane or its other qualities.

Army and Navy More Exacting than Airlines

JOHN G. LEE:—The single-engine transport mentioned seats nine passengers and the pilot. The operating-cost figures do not cover a long period, as the model has not been in service very long.

I hardly know how to answer the question as to the relative standing of military and commercial maintenance. I feel that the Army and the Navy are much more exacting to deal with in the factory than the commercial operators are, from which I infer that they are much more particular about what they get. I do not want to make trouble for the home factory, but I think the operators would get better products if their maintenance men were fussier.

The day when the commercial line got the rejects from the Army and the Navy has long since passed. Much more attention and care are given to the minor yet important details if purchasers insist that they be looked after. The factor that has penalized the commercial lines is the requiring of so much haste in building planes that not enough time is allowed for putting good features into them.

Airplanes are maintained on some airlines in a better way than they are on certain military fields, I think, although I should dislike to have to prove that point. The standards of maintenance on the airlines are very creditable, and in certain cases the experience of the commercial operators has become of more value than the military experience. I have reference to operations in Alaska and in the Canadian North, where commercial companies have been operating at temperatures much below any that our military services are consistently able to find. In cases of that sort we find that the commercial men have much better experience upon which to base design than the military services have.

A.C.C.A. Working on the Problem

WILLIAM E. BERCHTOLD[5]:—Mr. Warner's suggestion that the Air Transport Section of the Aeronautical Chamber of Commerce take up this matter of specification on maintenance needs is a good one. Mr. Lee's paper and Mr. Lockwood's very able articles that were published in *Aviation* provide a basic method for pursuing details of this subject. It looks like a difficult task, but it is not at all hopeless.

We have observed a real desire on the part of the manufacturers to do a good job, and certainly the operators are more awake than formerly to this problem of reducing maintenance costs. We have come to a point where it is necessary, if airline operation is to continue without financial loss, to definitely include these maintenance requirements.

CHAIRMAN MARSHALL:—Has not the Transport Section of the Chamber been for some time compiling a book of suggestions contributed by all the transport operators, which will be a standard of maintenance?

MR. BERCHTOLD:—That book has not been compiled so far particularly as a standard of maintenance to be used when specifying, but we have a forum for the exchange of information among the various airline operators on maintenance methods and short-cuts that can definitely reduce maintenance costs. We have kept that book as a compilation of confidential information for the members of the Transport Section of the Chamber and it is not available for general distribution, although some day it may be released. Certain definite recommendations were drawn up at our Dallas meeting last February and were unanimously approved by the operators, with the idea of passing them on to the engine, accessory and airplane manufacturers. Those for the engine manufacturers have already been sent out, and I am told that our New York office has received some replies suggesting ways of effecting some of the recommendations. We hope to do the same thing with the accessory manufacturers, particularly with those dealing with radio and the associated items.

S. PAUL JOHNSTON[6]:—I have just returned from a three-week's trip covering most of the major airlines east of the Rocky Mountains and, from the observations that I made, would like to confirm many of the statements and opinions that have been expressed here. Certainly a great need exists for more cooperation between manufacturers and operators than has obtained in the past. I think the time is coming when operators must write a very detailed specification for what they want.

[5] Manager, air transport section, Aeronautical Chamber of Commerce of America, Inc., New York City.

[6] Assistant editor, *Aviation*, New York City.

Legislative Regulation of Motor-Vehicles

Transportation Meeting Paper

By Thomas H. MacDonald[1] and J. T. Thompson[2]

AGREEMENT between designers of motor-transport vehicles and designers of highways is necessary, as the development of economical highway transport depends upon mutual adaptation of the vehicle and the road.

The chief obstacle to a solution of the problem is lack of uniformity of State laws and regulations governing the dimensions, weight, equipment and speed of vehicles.

The authors endeavor herein to present a composite picture of present thought as to what constitutes the most desirable recommendations for legislation. They present a set of recommendations prepared for consideration by the Committee on Highway Transport of the American Association of State Highway Officials and discuss briefly the reasons for each.

EFFICIENT development of highway transport in its more strictly engineering phases demands concord between the designers of the vehicle and the designers of the roadway. In addressing this paper to the Society of Automotive Engineers, we have endeavored to present a composite picture of modern thought upon highway design, insofar as it can be concretely defined at the moment, rather than to set forth individual opinion. Many blank places in our knowledge remain to be filled through painstaking research, and experience is disclosing many considerations besides the purely engineering factors. One fact, however, is certain: True economy of highway transport can result only from the mutual adaptation of the vehicle and the roadway.

For thousands of years the physical characteristics of vehicles and their operation have been regulated, in one form or another, by law. While it is a far cry from the bullock-drawn Roman cargo wagon lumbering through the provinces over pioneer roads to the present commercial motor-vehicles speeding on pneumatic tires over modern highways, the laws governing their activities have been aimed at the same basic question:

> What limits can the public permit the individual user so that commodities and persons may be transported quickly and most economically in the interest of the traveler, the producer, the transporter and the consumer without hurt to other users of the road, the taxpaying public or the highway itself?

Barriers Erected by Shortsighted Self-Interest

The problem today is a thousand times more complicated than ever before in history and demands not only judgment and the skillful use of scientific data, but also cooperation and a high degree of tolerance on the part of all who have a voice or an interest in its proper solution. The principal difficulty standing in the way of a reasonable solution of the engineering problem is not a lack of scientific data or of men capable of exercising the proper judgment in applying existing knowledge, but a lack of uniformity in our motor-vehicle laws. A close study of existing statutes forcefully impresses the wide and in most cases unfounded variations that exist from State to State. In the United States, we see, on one hand, 48 States linked with splendid highways and by the common bonds of language, tradition and commerce, and, on the other hand, the anomalous invisible barriers that self-interest, lack of foresight and lack of reciprocity have erected.

The sound principle of the American Constitution protecting interstate commerce is effectively attacked through the means of transportation.

To overcome this obstacle of non-uniformity in motor-vehicle regulations, several national organizations have from time to time issued recommendations and standard codes with the hope that these would serve to guide legislators and administrators throughout the Country. Among such organizations may be mentioned the National Conference on Street and Highway Safety, the Motor-Vehicle Conference Committee and the Committee on Highway Transport of the American Association of State Highway Officials. While their recommendations have not always been in exact agreement, common observations and the findings of engineering research made available to the memberships of all committees have tended to narrow disparities so that they are now approaching essential agreement.

In what follows, a set of recommendations prepared for the consideration of the Committee on Highway Transport of the American Association of State Highway Officials is discussed at some length. These recommendations concern the width, height and length of individual vehicles and possible combinations of motor-vehicles. They also include speed of operation, axle-load limitations and gross-weight restrictions. For each of these items the recommendation is given, followed by a discussion of the reasons underlying the proposal.

Just as it is impossible to conceive of the railroad engineer designing the trackage and road structures without due consideration of the use to be made of these facilities by the rolling stock, so is it apparent that the highway administrator cannot intelligently design a highway system without giving careful consideration to its probable use by motor transport. Accordingly, several important conferences have been held at which were present highway administrators, traffic officials and motor-vehicle manufacturers and operators.

The following recommendations and discussions thereof reflect to some extent the results of these conferences:

Width

Recommendation

No vehicle shall exceed a total outside width, including any load thereon, of 8 ft. except vehicles now in operation which, by reason of the substitution of balloon tires for other types of tire, exceed the above limit.

[1] Chief, Bureau of Public Roads, Department of Agriculture, City of Washington.

[2] Highway research specialist, division of tests, Bureau of Public Roads, Department of Agriculture, City of Washington, and professor of civil engineering, Johns Hopkins University, Baltimore.

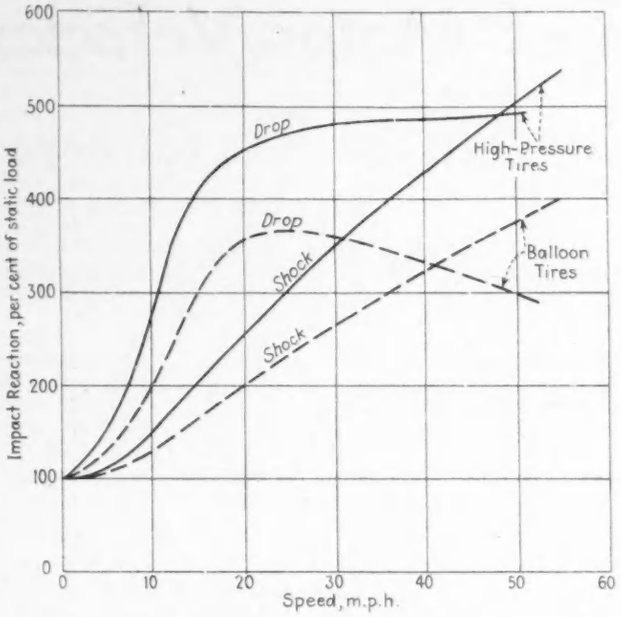

FIG. 1—IMPACT REACTIONS OF VEHICLES FITTED WITH HIGH-PRESSURE AND BALLOON TIRES

"Shock" Produced by Striking 1½ x 12-In. Rectangular Artificial Obstruction and "Drop" by Striking Pavement after Leaving Obstruction. The High Impacts Are Due to Exceptional Height of the Block

A study of the transverse distribution of traffic on paved highways[3] has demonstrated that trucks ordinarily move with outside wheels about 1½ ft. from the pavement edge. Virtually all of the States have built pavements of 18 ft. or less in width[4]. Also, the minimum curb-to-curb clearance recommended in current highway-bridge specifications[5] for two-way traffic is 18 ft. Since the maximum legal width of vehicles permitted in nearly all of the States is 8 ft., it is easily seen that any increase in this prevalent width should be discouraged. Two vehicles of 8-ft. width running on an 18-ft. roadway in the manner described would afford about 1½-ft. clearance when passing.

At present the great majority of the States have adopted the 8-ft. width. Only four States exceed this limit, while five have limiting widths less than 8 ft. The others subscribe to the 8-ft. width.

The recommended limit follows the Uniform Vehicle Code of the National Conference on Street and Highway Safety as well as that of the Motor-Vehicle Conference Committee.

Height

Recommendation

No vehicle unladen or with load shall exceed a height of 12 ft.

Need for a uniform height-recommendation is apparent. The height restriction in current State regulations ranges from 11 ft. to 14 ft. 6 in., while 12 States have no restrictions whatever on height. The wind-bracing requirements of through bridges and the economic desirability of small vertical clearances in grade-separation projects are important added reasons for height restrictions.

[3] See *Public Roads*, March, 1925, p. 1.
[4] See *Public Roads*, May, 1932, p. 47.
[5] See Standard Specifications for Highway Bridges, American Association of State Highway Officials, 1931, p. 167.
[6] See *Public Roads*, April, 1931, p. 21.

The recommendation follows the Uniform Vehicle Code of the National Conference on Street and Highway Safety.

Length

Recommendation

No single vehicle shall exceed a length of 35-ft. extreme over-all dimension including front and rear bumpers.

Combinations of vehicles shall consist of not more than two units and, when so combined, shall not exceed a total length of 65 ft.

A truck-tractor and semi-trailer shall be construed as constituting one unit of a combination of vehicles.

The need for a uniform regulation of the length of single and combined vehicles is made evident by the great divergence of existing regulations in this respect. Although all but eight States restrict the length of single vehicles, the legal lengths run from 26½ to 40 ft. As regards combinations of vehicles, all but seven of the States restrict the length, the restriction varying from 30 to 85 ft.

The suggested restriction of the length of a combination of vehicles to 65 ft. and the number of units of which the combination may consist, is based largely upon considerations of the safety and convenience of the motoring public.

Speed

Recommendation

(*a*) *Minimum speed.*—No motor-vehicle shall be unnecessarily driven at such a slow speed as to impede or block the normal and reasonable movement of traffic except when reduced speed is necessary for safe operation or when a vehicle or a combination of vehicles is necessarily or in compliance with law proceeding at reduced speed.

(*b*) *Maximum speed.*—No bus or truck shall be operated at a speed greater than 45 m.p.h. The operation of a passenger automobile at a speed greater than 45 m.p.h. shall constitute *prima facie* evidence of reckless driving.

Speed regulation must be viewed from two angles: (*a*) the convenience and safety of the motoring public and (*b*) the damage done to roads.

Minimum speed.—The fact is well known that much inconvenience and many accidents are caused by the slow driver. He dams up traffic behind him until the natural inclination to dart out of the traffic lane and hurry ahead in the face of oncoming traffic to a more favorable position often becomes too strong to resist. Many serious accidents are thus caused.

Maximum speed.—From the standpoint of safety it is obvious that excessive speeds must be curbed. From the road-damage viewpoint the fact has been known since the earliest days of impact investigation that the force applied to a pavement of given roughness by a wheel of a given static weight varies with speed. More recently[6] we have been able to segregate "shock" impact, caused by the wheel striking an obstruction, and "drop" impact, resulting from the subsequent dropping of the wheel upon the pavement. We know now that the shock type increases almost proportionately with speed, at least up to a speed of about 55 m.p.h., and that the drop type reaches its maximum at a speed of about 25 m.p.h. and then either decreases or fails to increase. This is shown in Fig. 1, from which we can also see that the shock impact does not reach the value obtained at relatively low speeds by the drop type until speeds around 50 m.p.h. are encountered.

In view of the foregoing, it is suggested that security from excessive impacts by motorcoaches and trucks mounted on pneumatic tires would result if the speed were limited to 45 m.p.h.

Axle Load

Recommendation

No vehicle shall be operated whose axle load, if equipped with high-pressure pneumatic tires, exceeds 16,000 lb. or whose axle load, if equipped with low-pressure, or balloon, tires, exceeds 18,000 lb.

These limitations are recommended as the minimum maximum limits to be imposed for main roads in any State, but should not be construed as inhibiting heavier axle loads if any State desires a higher wheel-load limit.

An axle load shall be defined as the total load on all wheels whose centers may be included between two parallel transverse vertical planes 40 in. apart.

The problem of fixing limiting loads is not a simple one. Opposed to the natural desire of the highway administrator to protect existing construction against damage and to expand his system as far as possible with available funds, is the desire of the highway transport business to operate with loads that it considers efficient and economical. The administrator naturally favors lighter loads, whereas some operators favor heavier wheel loads. The need of reconciling these viewpoints is obvious, and the desirability of uniformity in this particular is apparent, especially since in the existing State regulations permissible wheel loads range from 4000 to 17,200 lb. It should be stated, however, that 31 States have adopted either the 8000 or 9000-lb. maximum wheel load or some intermediate value.

The solution of the problem lies in being able to predict with a fair degree of accuracy the effect of wheel loads upon pavements. This is a very involved problem.

The force applied to the pavement is a function of static wheel load combined with impact, the latter being a function of vehicle speed, road roughness and vehicle characteristics, particularly tire equipment. The effect of this force depends upon the area over which it is distributed by the tire, pavement dimensions and materials, inertia effects of both pavement and subgrade and subgrade resistance.

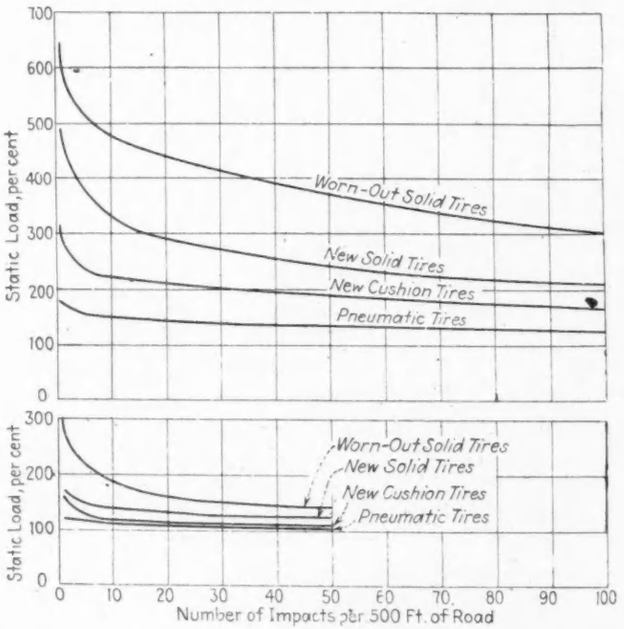

FIG. 2—INFLUENCE OF SPEED, ROAD ROUGHNESS AND TIRE EQUIPMENT IN PRODUCING AN APPLIED FORCE IN EXCESS OF STATIC WHEEL LOAD

(Upper Chart) Applied Force on Rough Stone-Block Pavement; (Lower Chart), on Smooth Concrete Pavement. Conditions: Truck, 2 Ton; Tires, Rated Sizes, Dual; Wheel Load, Tire Capacity; Speed, 12 M.P.H.

TABLE 1—COMPARISON OF STRESSES IN 9-6-9-IN. CONCRETE PAVEMENT, WITH NO IMPACT AND WITH IMPACT[a], CAUSED BY 8000-LB. WHEEL LOAD ON HIGH-PRESSURE TIRES AND 9000-LB. WHEEL LOAD ON LOW-PRESSURE TIRES

Wheel Load, lb.	8,000		9,000	
Tire Equipment	40 x 8 H-P Dual		10.5 x 20 L-P Dual	
	No Impact	With Impact	No Impact	With Impact
Gross Area of Tire Contact, sq. in.[b]	170	200	206	239
Value of Radius, in.[c]				
Interior	7.35	7.98	8.10	8.73
Edge	10.40	11.29	11.45	12.34
Fiber Stress, lb. per sq. in.[d]				
Interior	260	367	280	351
Edge	170	175	233	218

[a] Trucks traveling at 30 m.p.h. on pavement of average roughness. Both trucks with specified tire equipment give a total impact reaction of 11,700 lb.

[b] Computed by adding contact area of one tire to length of area times center-to-center spacing of dual tires.

[c] For interior stresses use radius of circle equivalent in area to gross area of contact. For edge stresses use that equivalent to semicircle.

[d] Values picked from Fig. 4 and multiplied by static or impact wheel load divided by 10,000.

Effect of Impact Forces on Pavement

In the last 10 or 12 years a great mass of valuable information has been collected concerning the cause, nature and effect of forces due to moving vehicles. The principal sources of this information are the Bates road tests and subsequent work on impact by the State of Illinois, the Pittsburg, Calif., tests and the researches of the Bureau of Public Roads.

As a result of this work we are able to predict within reasonably narrow limits what actual force will be applied to a road surface by a wheel of given static weight if the speed, surface roughness and tire and vehicle characteristics are defined. The influence of speed, roughness and tire equipment in producing an applied force in excess of the static weight of the wheel is typically illustrated by Figs. 1 and 2, which are self-explanatory.

But mere evaluation of the applied forces does not solve the problem; the *effect* of these forces is the important factor. So far as rigid-slab pavements are concerned, we have a workable and mathematically sound theory of stress prediction, and, were the slab perfectly smooth and wheels perfectly circular, a condition that gives rise to no impact increment, we should be able to predict stresses in such a pavement with a fair degree of accuracy. The static weight of the wheel would be the determining factor under these circumstances. Indeed, the rapid elimination of the solid tire and the improvement of pavement specifications and finishing methods have tended to reduce wheel roughness on the one hand and surface roughness on the other so that we approach the ideal condition. Roughness of a serious degree will be found only at certain places such, for example, as where two adjacent rigid road-slabs have separated vertically. These places of extreme roughness will always cause impacts of a destructive nature and will call for maintenance or replacement as such limited areas develop. By and large, however, design and stress prediction will be governed by a degree of roughness that is much less serious.

In the case of surfaces that are not inherently smooth and of non-rigid types that are susceptible to increasing roughness under traffic, the resulting impact increments must be considered.

Following the adoption of the thickened-edge pavement by the State of Illinois, most of the States now

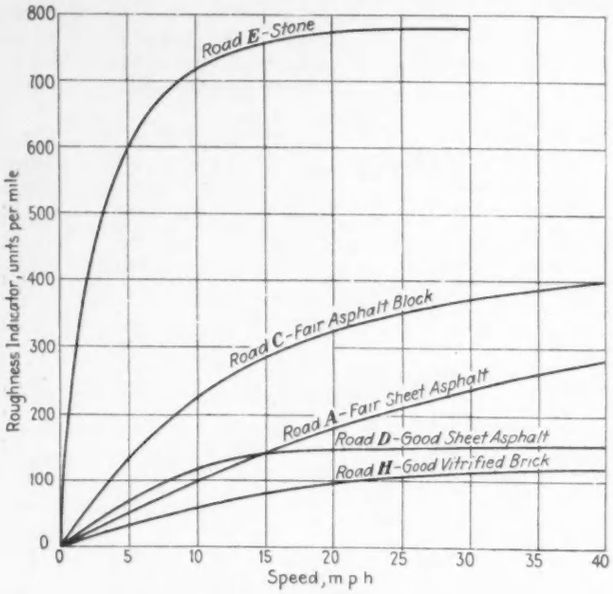

FIG. 3—CALIBRATION OF ROUGHNESS OF ROAD TEST-SECTIONS WITH RELATIVE-ROUGHNESS INDICATOR

construct concrete pavements with a thickened edge. The most common designs call for either a 9-6-9-in. or a 9-7-9-in. cross-section. As previously mentioned, the most common limitations upon maximum axle loads are 16,000 or 18,000 lb., corresponding to wheel loads of 8000 or 9000 lb. Table 1 shows the relative effect upon the 9-6-9-in. concrete pavement of the 8000-lb. wheel equipped with high-pressure and the 9000-lb. wheel equipped with low-pressure or balloon tires. The no-impact figures are based upon the assumption that the surface is sufficiently smooth to ignore impact; those with impact, on a fair degree of roughness, say that corresponding to the fair sheet-asphalt pavement designated Road A in Fig. 3, and a speed of 30 m.p.h. The stresses in all cases were picked from Fig. 4, which shows the marked influence that area of tire contact has upon stress. It should be stated also that the subgrade coefficient used in these computations corresponds to a poor subgrade, one which will permit 1-in. penetration of a 1 x 1-in. block when loaded with 50 lb.

Because this comparison shows that no appreciable difference exists between the effect of the 8000-lb. wheels fitted with high-pressure tires and the 9000-lb. wheels fitted with low-pressure tires, the conclusion reached is that it is proper to make the recommendation to which this discussion refers.

Balloon Tires Reduce Road Stress

If, as a result of the Bates road tests, the 8000-lb. wheel load was generally deemed safe for the 9-6-9-in. pavement, we certainly can accept the 9000-lb. wheel load with balloon tires today. In nearly all legislation fixing the 8000-lb. limitation, little or no restriction was placed upon the type of tire equipment. As a result, 8000 lb. per wheel was rather generally permitted on solid tires. Indeed, this combination is now legal in many States. If, as has been shown, the 9000-lb. wheel load on balloon tires causes pavement stress essentially equal to that caused by the 8000-lb. wheel load on high-pressure pneumatics, obviously the 8000-lb. load on solid tires will cause much higher stresses.

A direct comparison of a 9400-lb. wheel load on balloon tires with one of 8000-lb. on new solids is offered. Using observed impact reactions with 40 x 6-in. new solid dual tires on a truck chassis operated at 12 m.p.h. and 12.50 x 20 dual balloon tires on a motorcoach chassis operated at 40 m.p.h., and the contact areas occasioned by a 30 x 1½-in. inclined-plane obstruction, the edge stress[7] caused in a 9-6-9-in. pavement by the 9400-lb. balloon-tire combination is 450 lb. per sq. in., whereas that caused by the 8000-lb. new solid-tire combination is 1140 lb. per sq. in.

It should be noted that the 1½-in. obstruction results in an unusually severe impact. Fortunately, however, the solid tire is fast disappearing. The number of new trucks equipped with solid tires has dropped from 25 per cent of the total number produced in 1922 to 3.1 per cent in 1931.

In addition to fixing the maximum legal wheel load, the arrangement of axles also should be considered. Experiments[8] have definitely shown that, where a given load is equally distributed over two axles, the stresses in the pavement are not greater than if half the same load were carried on one axle, provided that the two axles are at least 36 in. apart. In recognition of this fact, an axle load should be defined as the total load on all wheels whose centers may be included between two parallel transverse vertical planes 40 in. apart.

Gross Weight

Recommendation

Subject to the recommended axle loads, no vehicle shall be operated whose total gross weight, with load, exceeds that given by the formula

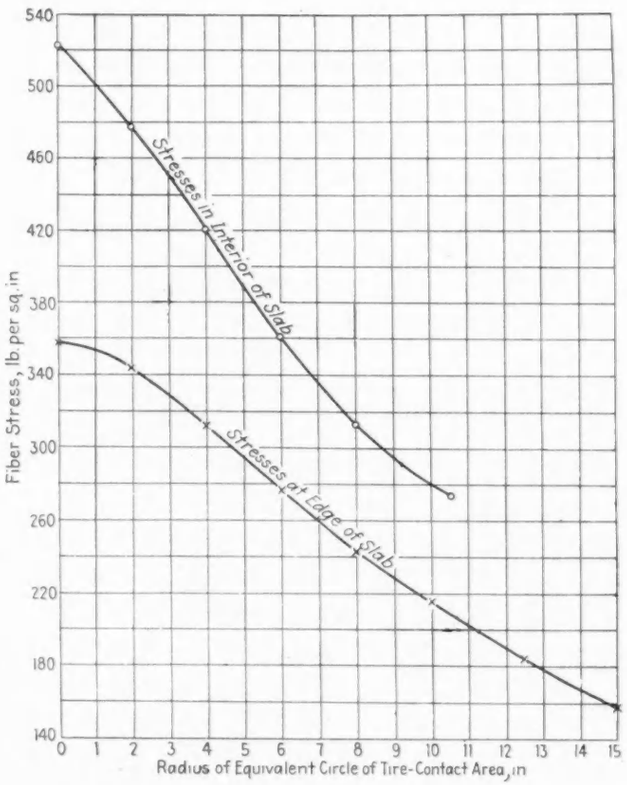

FIG. 4—INFLUENCE OF TIRE-CONTACT AREA ON INTERIOR AND EDGE STRESSES IN A 9-6-9-IN. CONCRETE SLAB

The Stresses Are Plotted for a 10,000-Lb. Wheel Load but Are Directly Proportional to the Load. They Are Computed by Westgaard's Formula with $E=3\times10^6$, $\mu=0.15$, $\kappa=50$.

[7] Stresses computed by Westgaard's formula.

[8] See *Public Roads*, October, 1925, p. 165; also Report No. 31-8, State of Illinois, Department of Public Works and Buildings, Division of Highways, 1931, p. 7.

$$W = C(L + 40)$$

where

C = a coefficient to be determined by the individual States
L = distance between the first and last axles of a vehicle or combination of vehicles in feet
W = total gross weight, with load, in pounds

A value of 800 is recommended for C as the lowest which should be imposed, but this should not be construed as inhibiting greater values.

The methods in force at present for limiting gross vehicle weights are entirely lacking in uniformity. These methods generally are based upon such factors as number of wheels or axles, type of tire and number of vehicles in combination. A study of a recent compilation of restrictions reveals that for four-wheel vehicles the permitted gross weights varied from 8000 to 36,000 lb., for the six-wheel type from 16,000 to 48,000 lb., while for combinations of vehicles the disparity was even more pronounced[9].

An exhaustive study of the effect of various combinations of current commercial vehicles made by the Bureau of Public Roads has shown that the legal gross weight can logically be expressed by the suggested formula $W = C\ (L + 40)$. It is assumed that each State will have to fix for itself the value of the coefficient C after a study of its existing highway structures and in the light of future planning.

The recommended value $C = 800$ is based upon the fact that it will permit the safe operation of

(1) All four-wheel vehicles of current design upon H-10 bridges[10].
(2) All six-wheel vehicles of current design upon H-15 bridges.
(3) The commonest types of truck-tractor and semi-trailer, that is four-wheel or six-wheel tractor and two-wheel semi-trailer, upon H-15 bridges. Unusual combinations, such as the six-wheel tractor and four-wheel semi-trailer, are excluded except when operated with reduced wheel loads.
(4) The commonest type of truck-tractor, semi-trailer and one full trailer upon H-15 bridges. This type of combination calls for four-wheel tractor, two-wheel semi-trailer and four-wheel full trailer. Unusual combinations, such as the six-wheel tractor, four-wheel semi-trailer and six-wheel full trailer, may be operated only with drastically reduced wheel loads.

In conclusion, the crying need today is for uniformity in State laws regulating motor-transport activity. Many phases of this activity are beyond the scope of this paper and in them the same lack of uniformity exists as in the items touched upon here. To obtain uniformity we must first have a set of regulations or a code to which all interested parties can reasonably adhere; a code based upon experience, observation and research data. For the limited items involved, it is believed and hoped that the recommendations discussed in this paper may provide a code that is reasonable and that may be generally accepted in view of the basic necessity to the future of highway transport.

[9] See *Commercial Car Journal*, July, 1932.

[10] See Standard Specifications for Highway Bridges, American Association of State Highway Officials, 1931.

Justifying Motor-Truck Transportation

IF one listens, without critical analysis, to the complaints of our rail competitors, no justification whatever for the existence of motor-truck transportation would seem to exist. In spite of the fact that in 1929 the Hoover Committee on Recent Economic Changes found little to fear from truck transportation, the cry issuing from the mouths of railroad officials for extreme regulation of trucks indicates that we are a growing factor in the transport industry.

We cannot grow unless we provide a service that is needed, at a time when it is needed and at a price that the public is willing to pay. I take it that the increasing use by the business public of motor-truck transportation is undeniable evidence that we are providing a service for which a need exists and that therefore we are entitled to operate and to continue to expand our operations.

We public-carrier truck operators should go even farther to justify our existence. I believe that, as we accumulate knowledge, we shall find commodities that do not now move in quantity either by truck or rail but that will so move when we are able to furnish transport facilities that make the movement of such commodities not only possible but profitable to us all.

No well-defined policy of regulation has been expressed as yet. We do not want severe regulation, yet we believe that regulation is not only just but desirable. The arguments for regulation are based on our use of public highways and on the assumption that motor-trucks inflict greater wear and tear on public highways than do passenger-cars. On a basis of comparative weight this could be true, but only in those cases where the public highways are built with stress factors of safety that did not anticipate the use of heavy trucks. On the other hand, a study of any given stretch of road used by truck systems probably would show that the passenger-car miles times passenger-car average weight is greatly in excess of the comparable figures for public-carrier trucks.

We of the truck-transport industry cannot and should not try to avoid taxation and regulation that are intended to charge us with our proportionate share of the use of public roads and to prevent our industry from engaging in practices contrary to sound public policy. But we must, and will, fight any taxation and regulation that spring from the desire of our rail competitors to throttle our progress in a field of service that they admittedly cannot handle at rates that return a profit.

Our costs of sales acquisition are as yet a problem on which we have little information. Our costs will be adversely affected by every loss we pay arising out of the failure of our equipment or personnel. To the extent that equipment failure contributes to poor service, the automotive engineers can be of assistance in locating and eliminating basic causes for such failures, thereby reducing our sales cost.

The public-carrier truck organization is able to build its selling force to the end that our customers shall experience a more personal, friendly contact than they experience from other methods of transport.

Our insurance costs are much too high. Possibly, since the trucking business is in an embryonic stage, insurance companies have not been able to accumulate sufficient experience to allow of more equitable premiums. On the other hand, our equipment is not guarded by mechanisms that will show, as by the speedometer, exactly when and at what point doors were opened. Equipment of this kind would be useful in checking thefts in transit. Losses due to breakage in handling are of a more personal nature and more certainly to be solved by intelligent training of personnel than by any other means.

Our tractor and cargo-equipment operating costs are too high. We are watching with interest the application of the Diesel engine to truck tractors. Successfully adapted, it means great savings in fuel costs and in the life of tractor engines.

I feel that the automotive engineers should leave no stone unturned in their efforts to provide us with equipment that will enable us to continue to justify our existence as a part of the best transportation system of which the world has knowledge.—From a Kansas City Section paper by C. E. Lang, president, Missouri Motors Distributing Corp.

Effect of Legislation on Motor-Vehicle Design and Operation

By Pierre Schon[1]

Transportation Meeting Paper

EXISTING legal restrictions prevent the public from deriving the utmost benefits from the progress made in transport-vehicle and highway engineering. Legislative regulation has not yet affected the design of passenger automobiles in this Country, but curtailment of usage is evident in those States where gasoline taxes have reached exorbitant levels. The design and operation of motorcoaches, trucks and trailers has been affected, and the trend of motor-vehicle legislation presents a problem that is more acute than ever before in the history of the industry.

We have 49 different sets of State and District regulations, each differing in some ways from the others, most seriously as regards size and weight. If uniform regulations could be put into effect in all States, design and operating practices would be simplified and lower manufacturing and operating costs effected.

The author discusses in detail the effect on design of the limitations of over-all width, height, length of single vehicles and combinations of vehicles, number of units in a combination, speed and gross weight and weight per axle.

While agreeing with some of the recommendations made to the American Association of State Highway Officials by the Bureau of Public Roads, practical objections to some others are pointed out.

THE trend of legislative regulation and its effect on the design and operation of motor-vehicles present a problem for manufacturers and operators that is more acute now than at any time in the history of the automotive industry. Regardless of engineering research and development in highway-transport vehicles from the standpoint of performance ability and reduction in the cost of operation and regardless of the great achievements in improving the methods and reducing the cost of building good highways, legal regulations prevent the general public from deriving the utmost benefits from the progress made by these two branches of engineering.

While legislative regulations have not as yet affected the design of passenger-cars, curtailment of usage is already evident in those States where gasoline taxes have reached exorbitant levels as is shown in Table 1. Vehicle registrations in the United States show that last year 711,462 fewer passenger-cars were in use than in 1930. No doubt our depression period has been partly responsible for this loss in registrations, yet official statistics show that in those States imposing a low tax on gasoline, vehicle registrations actually increased regardless of the depression, while in the States having higher gasoline taxes, the registrations gradually decreased.

High cost of gasoline has exerted a strong influence on the design of passenger-cars in European countries and, while the average tax on gasoline for the United States at this time is approximately 5 1/7 cents per gal., in certain localities this particular tax has become an unbearable burden to many people; for instance, in Mobile, Ala., motorists pay a tax on gasoline amounting to 11 cents per gal., divided as follows: Federal, 1 cent; State, 5 cents; county, 3 cents; city, 2 cents. The frantic search by our government bodies for new sources of taxation may eventually affect the design of passenger-cars in this Country, the same as high taxes have forced European manufacturers to produce light-weight cars giving higher gasoline mileage.

TABLE 1—DECREASE IN MOTOR-VEHICLE REGISTRATIONS IN 1931 COMPARED WITH 1930 IN STATES HAVING THE HIGHER GASOLINE TAXES

State Gasoline Tax, Cents	Vehicle Registrations	
	Increase, Per Cent	Decrease, Per Cent
2	1.8	
3		1.0
4		2.3
5		7.5
6		13.2

We are concerned here primarily, however, with existing regulations applying to heavy motor-vehicles and their effects on the design and operation of motorcoaches, trucks and trailers. While this is not the time nor the place to discuss the motives prompting the proposals of drastic restrictive regulations with which our State legislatures are being flooded, it is an opportune time to state that, unless the activities of the propagandists behind this nation-wide movement are curbed, the year 1933, with 44 State legislatures scheduled to be in session, will bring about the most critical situation ever faced by the automotive industry.

Based on legislative enactments during 1932, no one can foretell to what extent the juggling and shuffling of State motor-vehicle laws will affect our industry, both manufacturers and operators. Bills for regulating highway transport to the extent of confiscation and elimi-

[1] M.S.A.E.—Sales engineer, General Motors Truck Co., Pontiac, Mich.

nation of a competitor in the transportation business are being written by lawyers for special interests, and these bills are being supported by the organized forces of 21 labor unions which are actively engaged in circulating petitions to present in every State legislature scheduled to meet next winter. Regardless of facts and figures established by the Bureau of Public Roads relative to highway construction and of facts developed showing the value of highway transportation to our industries, our farmers and the consumer, the proponents of drastic regulation have so far exhibited the utmost disregard for uniformity of regulations of size and weight. As the actual picture stands today, we have 49 different sets of State and District regulations differing in some respects from those of neighboring States.

If a uniform code controlling size and weight of vehicles, along the lines recommended by the American Association of State Highway Officials, could be put into effect in all States, existing design and operating practices would be simplified, resulting in lower manufacturing costs and greater economies in operation.

Complications Produced by Width Regulations

In dealing with legal limitations prescribing maximum sizes of vehicle, we find that a fair degree of uniformity is apparent in the width regulations, as shown in Fig. 1, a width limit of 96 in. being in effect in a majority of the States. Massachusetts has the most practical law, with a basic 96-in. vehicle limit but permitting 102-in. width from outside to outside of dual pneumatic tires.

Enforcement of the 96-in. law frequently results in complications for the operator, regardless of the fact that the manufacturing limits comply with the legal regulation. This is because of the peculiarity of a balloon tire growing with age and also the low air pressure allowing bulging of the tire at the bottom. Unless a provision is made in the regulations that a slight increase over the legal limit caused by inflation, deflection, wear or bulging of balloon tires shall not be considered an infraction of the law, the manufacturers of

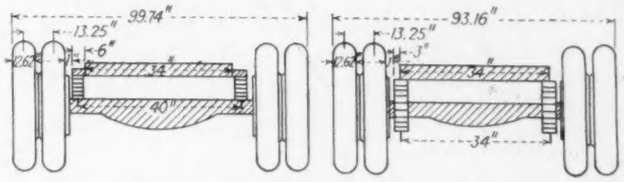

FIG. 2—CHANGES IN VEHICLE DESIGN NECESSITATED IF THE BALLOON TIRES OF THE LARGER SIZES ARE TO BE USED AND THE WIDTH RESTRICTION OF 96 IN. IS STRICTLY ENFORCED

(Left) Truck Rear Axle with Outboard Spring-Mounting and 11.50/20 Dual Tires. (Right) Under-Frame Underslung Spring-Mounting and 11.50/20 Dual Tires

motorcoaches and trucks will be compelled to provide a maximum-width limit of less than 96 in. on new vehicles when measured from outside to outside of balloon tires.

Important changes in vehicle design are necessary to permit dualling the larger sizes of balloon tires if the 96-in. limit on width is strictly enforced. We must either reduce the width of the frame to impractical dimensions or locate the rear springs partly under the frame, as in Fig. 2. If we move the springs to an underframe position and the top of the frame is to be kept within a practical height from the ground, the rear springs must be placed under the axle. In any event, this change in design requires a special axle-housing design and reduction of the distance from center to center of the spring saddles from 40 in. to 34 in., decreasing the stability of the vehicle in relative proportion. Changes in rear-spring suspension brackets for attachment to the frame also must be made so as to provide adequate vertical clearance between the frame and the axle housing when the springs are deflected.

Statistics compiled by the Department of Commerce definitely indicate that the width of vehicles does not affect accidents or highway hazards. Florida, with an 84-in. limit, shows a much higher rate of motor-vehicle accidents, namely 34.4 per 100,000 population, than does Rhode Island, with an accident record of 16.7 and a

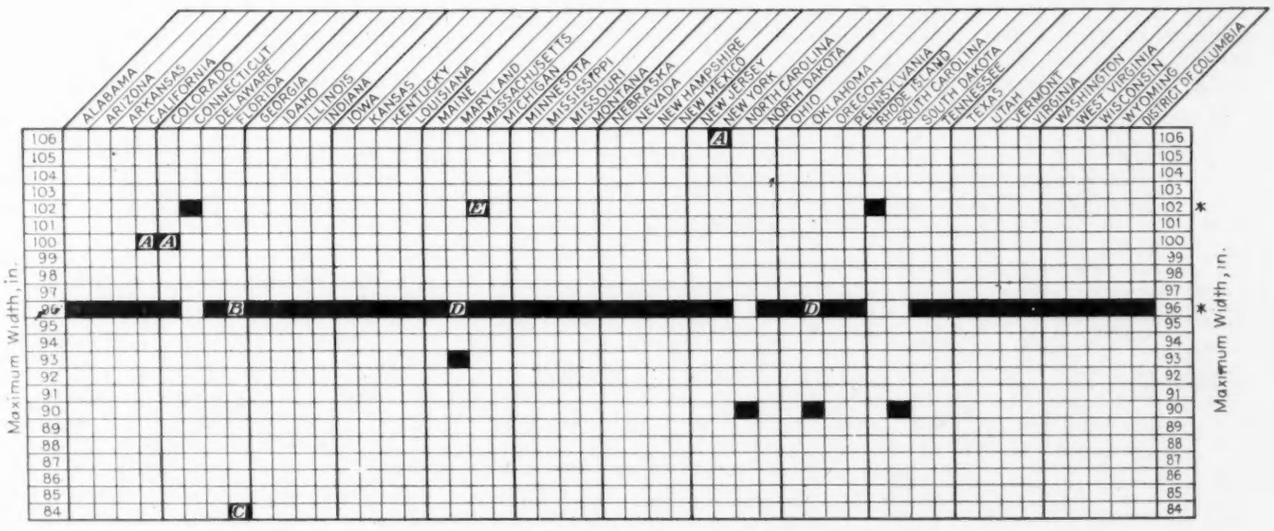

FIG. 1—INTERPRETATION OF WIDTH REGULATIONS AS OF OCT. 1, 1932

A—To Facilitate Change-Over from Solid to Pneumatic Tires. B—Contract and Common Carriers. C—Private Operators. D—By Special Permit. E—Dual Pneumatic Tires

BASIC LAWS

96-In. Width or More		Less than 96 In.	
No. of States	Width, In.	No. of States	Width, In.
2	102	1	93
42	96	3	90
		1	84

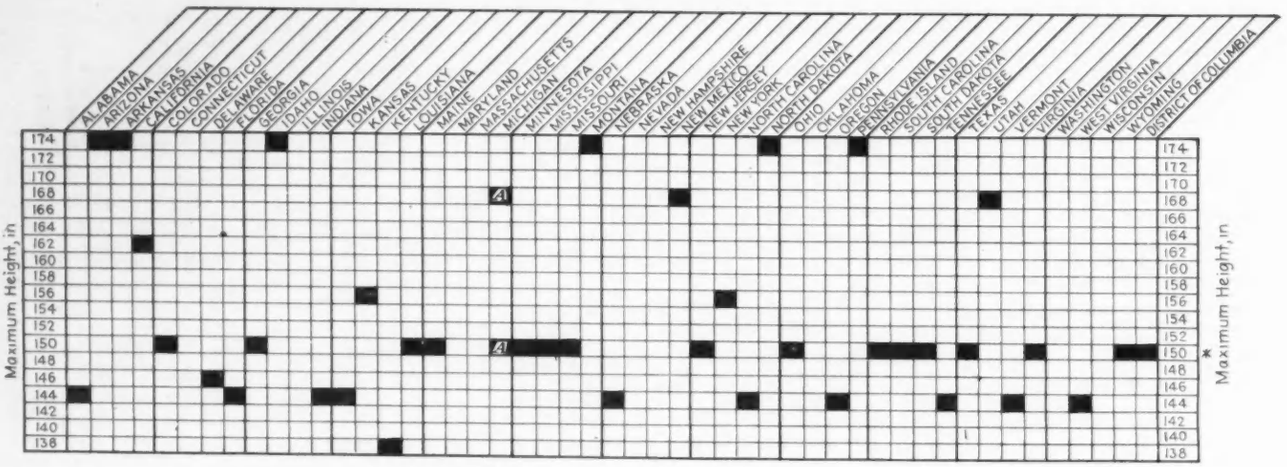

FIG. 3—INTERPRETATION OF HEIGHT REGULATIONS AS OF OCT. 1, 1932

A—Two Conflicting Laws in Effect

BASIC LAWS

12½ Ft. or More		Less than 12½ Ft.	
No. of States	Height, In.	No. of States	Height, In.
9	No Regulation	1	146
6	174	10	144
3	168	1	138
1	162		
2	156		
17	150		

width allowance of 102 in. Accidents in individual Florida cities as compared with Bridgeport, Conn., also show higher ratios of vehicle accidents; for example: Miami, 52.4; Jacksonville, 42.5; and Bridgeport, 40.1. Connecticut also allows 102 in. width.

The Massachusetts law is the most practical as regards design and operation and is in line with the rapid progress made in the general use of the balloon tire. The Massachusetts width law provides as follows:

> The 96-in. limit may be exceeded by the lateral projection of pneumatic tires beyond the rims of the wheels for such a distance on either side of the vehicle as will not increase its outside width above 102 in.

Wide variation exists in height restrictions, as revealed by Fig. 3, but the majority of States permit 12½ ft. or more, with only 12 States having laws that limit vehicle height to less than 12½ ft. or 150 in.

TABLE 2—LENGTH REGULATIONS FOR TRACTOR AND SEMI-TRAILER

State	Considered As	Legal Length Limits Semi-Trailer Only, Ft.	Total Length, Ft.
Illinois	1 Unit	None	35
Indiana	2 Units	35	40
Ohio	2 Units	50	None
Michigan	2 Units	None	60
Kentucky	1 Unit	None	30

While height regulations have not yet appreciably affected the design of vehicles, the precedent estab-

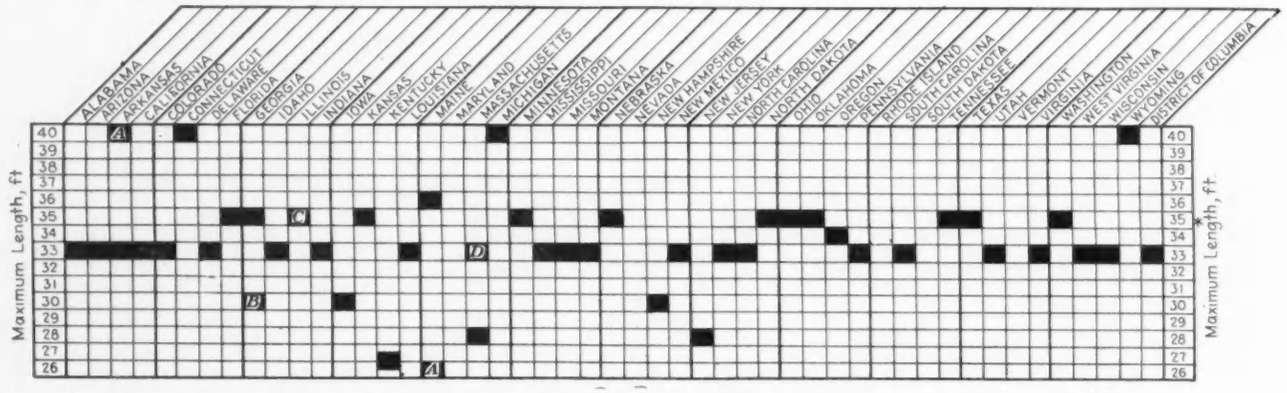

FIG. 4—INTERPRETATION OF SINGLE-VEHICLE-LENGTH REGULATIONS AS OF OCT. 1, 1932

A—Trailer or Semi-Trailer. B—Vehicles Privately Owned. C—Tractor and Semi-Trailer Considered as One Unit. D—In Localities Designated by the Department of Public Works

BASIC LAWS

35-Ft. Length or More		Less than 35 Ft.	
No. of States	Length, Ft.	No. of States	Length, Ft.
4	No Regulation	23	33
3	40	3	30
1	36	2	28
12	35	1	26½

lished in Kentucky this year limiting the maximum height to 11½ ft. and even more stringent regulations in prospect for 1933, may force us to adopt drop frames and make other important design changes for certain vocations to keep the over-all loading height within the law.

Since most of the States allow heights greater than 12 ft. and certain types of specialized vocational vehicles require a height of 12½ ft., a height limitation of 12½ ft. or 150 in. obviously would be a more desirable standard to adopt in a uniform code. Attention is also called to the fact that certain types of road-building machinery transported to and from highway-construction jobs on trailers of special design require a slightly greater clearance under bridges than the 12-ft. height recommended in the National Safety Code.

Difficulties Presented by Length-Limit Variation

A 35-ft. length, as recommended in the proposed uniform code, is entirely adequate and practical for a single-vehicle unit. Present regulations show a wide variation and urgent need exists for arriving at a standard uniform length limit throughout the United States. With limitations ranging from 26½ up to 40 ft., as in Fig. 4, vehicles used for interstate transportation must necessarily be held within the minimum length. Up to the present, length restrictions for single vehicles have not seriously affected the design of single vehicles, but our industry is on the threshold of radical design changes of the single unit for use in combinations of vehicles.

Telescoping frames, adjustable wheelbases and accordion-type bodies have been humorously suggested as solutions of the problem of enabling interstate operators to cross our State boundary lines and comply with existing and constantly changing State laws.

TABLE 3—STOPPING DISTANCES REQUIRED BY CONNECTICUT LAW

Speed, M.P.H.	Stopping Distance, Ft. — Service Brakes	Emergency Brakes	Emergency Brakes on Trucks over 2-Ton Capacity
20	40	60	80
25	70		
30	100		
35	140		
40	175		

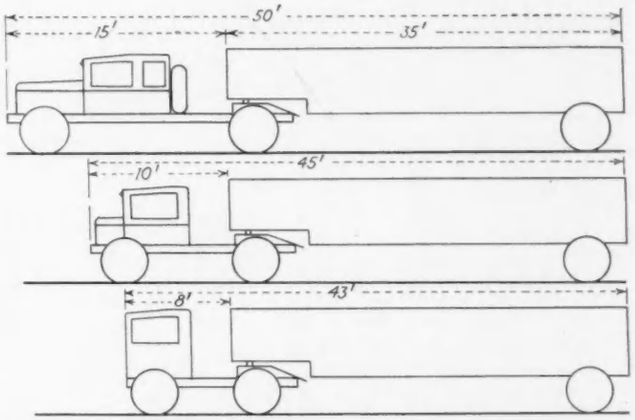

FIG. 5—HOW A SEMI-TRAILER LENGTH OF 35 FT. AFFECTS LENGTH OF A TRACTOR AND SEMI-TRAILER COMBINATION

Even with the Cab Built over the Tractor Engine, as in the Bottom Drawing, the Over-All Length Is 43 Ft.

The first step necessary to clarify the status of the most commonly used combination of vehicles is to determine whether a tractor and semi-trailer shall be regarded as a single unit or a combination of two units. A typical example of impractical regulations in five neighboring States is found in the laws of Illinois, Indiana, Michigan, Ohio and Kentucky. Whereas Illinois and Kentucky classify a tractor and semi-trailer as a single unit, in the other three States they are held to be a combination of two units, as shown in Table 2.

For registration and licensing purposes, a tractor and semi-trailer should be classified as two units. In transporting certain classes of bulky light-weight materials, where loading space is an important factor, the basic 35-ft. length recommended for the single vehicle is also highly desirable and practical for controlling the length of the semi-trailer.

Various design changes have been made to fit a tractor

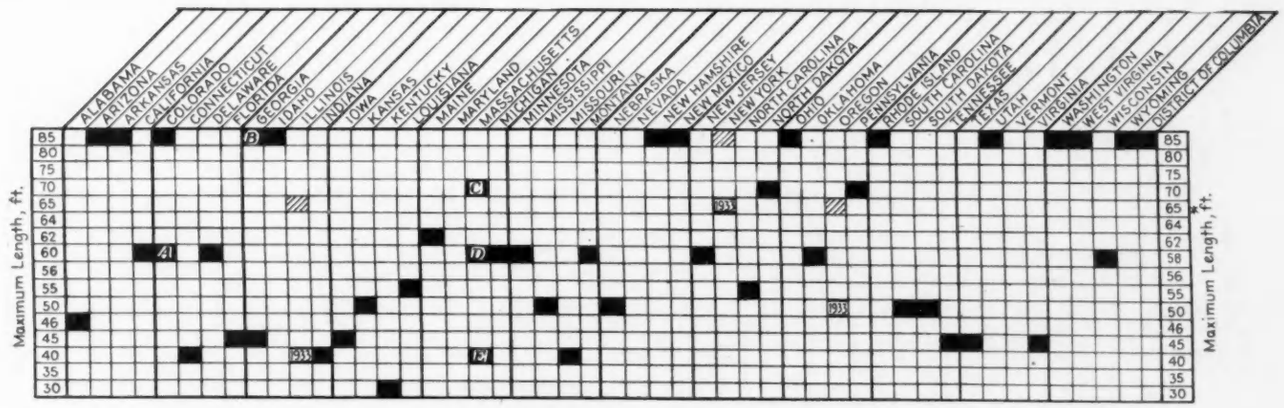

FIG. 6—INTERPRETATION OF LENGTH REGULATIONS FOR VEHICLE COMBINATIONS, AS OF OCT. 1, 1932

A—Combination Length on Mountain Roads. B—Vehicles Privately Owned. C—In Localities Designated by the Department of Public Works. D—Truck and Four-Wheel Trailer. E—Tractor and Semi-Trailer

BASIC LAWS

60-Ft. Length or More — No. of States	Length, Ft.	Less than 60 Ft. — No. of States	Length, Ft.
3	No Regulation	2	55
14	85	5	50
3	70	1	46
2	65	6	45
1	62	3	40
8	60	1	30

and semi-trailer to the length limits of different State laws, but no standard design practice can be established until the legal situation becomes more definitely clarified. Operators demand maximum loading space within legal limits. With the 35-ft. single-unit basic length, the additional length for the tractor obviously brings the over-all length of the conventional-type tractor-semi-trailer combination to the minimum of 45 ft. (See Fig. 5.) This is increased to 50 ft. where a sleeper cab and a spare tire back of the cab increase the length of the tractor ahead of the semi-trailer body. Even with reduced cab length, obtained by placing the cab over the engine, holding the over-all length of a tractor and semi-trailer within 40 ft. is impossible if the basic 35-ft. single-unit length is applied to the semi-trailer. A limit of 45 ft. is more practical for a tractor and semi-trailer combination.

A great variation also is apparent in legal limitations of length for combinations of vehicles, the restriction ranging from 30 to 85 ft. for the different States, as shown in Fig. 6. A 65-ft. maximum-length limit, as recommended for the proposed uniform code, is entirely practical for all ordinary hauling requirements. No truck operation of which we know requires a length of 85 ft. except for hauling steel girders, timbers, telephone poles or such articles as cannot reasonably be transported without being dismantled. To cover such exceptions, provision should be made in the laws for a special annual permit for one or more vehicles operated by the same applicant.

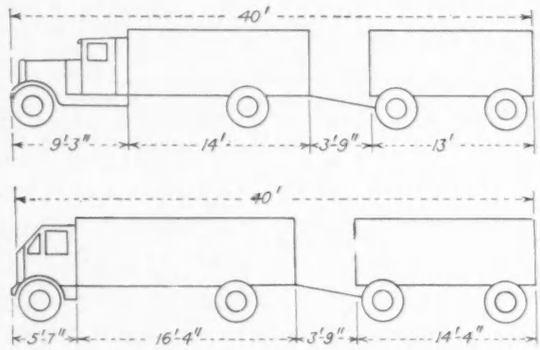

FIG. 7—DESIGN CHANGES IN TRUCK CAB TO MEET 40-FT. OVER-ALL-LENGTH LAW

To Provide Sufficient Body Load-Space for Furniture and General Merchandise in Truck and Trailer, the Cab Must Be Built over the Engine

Last year, Indiana and Illinois passed laws limiting the length of a vehicle combination to 40 ft. In attempting to meet the legal limitations in these two important truck markets, our first problem consists of providing adequate loading space for a truck and four-wheel trailer within the maximum over-all length of 40 ft. Assuming a distance of 9 ft. 3 in. from bumper to back of cab, plus 3 ft. 9 in. of space required between the two bodies, as illustrated in the upper view of Fig. 7, a total of only 27 ft. remains for useful body load-space—14 ft. for the truck and 13 ft. for the trailer body. These body lengths provide sufficient load space for tank trucks or hauling heavy materials such as coal, gravel and cement but obviously are too short to meet the practical loading requirements for furniture moving and transporting general merchandise and for many other vocations.

To meet this situation, truck manufacturers are compelled to make changes as illustrated in the lower view of Fig. 7, locating the cab over the engine within a distance of only 5 ft. 7 in. from bumper to back of cab. This allows a body length of 16 ft. 4 in. for the truck and 14 ft. 4 in. for the trailer within the over-all legal limit of 40 ft. The design changes involve the following major problems:

(1) Shortening distance between front bumper and back of cab
(2) Increasing carrying capacity of front axle and front springs
(3) Changes in steering-gear mounting
(4) Changes in control levers
(5) Development of a new cab

This new design not only results in a gain in body load-space but provides for a better load distribution utilizing the full carrying capacity of the front tires.

Number of Vehicles in a Combination

With the modern design of trailer, in which side-swaying has been reduced to the minimum, no valid reason can be advanced for considering three units an excessive number. Three vehicles coupled together have a shorter turning radius than two long vehicles within the same over-all combination length. In certain vocations the splitting of the load into more than two units is highly desirable, as in the hauling of milk from farms to the city. For instance, in Michigan an average of 2,000,000 lb. of milk per day is brought into the City of Detroit by vehicle combinations of one truck and two trailers. (See Fig. 8.) This milk is hauled from territories that are not served by railroads or any other means of transportation. The truck pulling two empty four-wheel trailers on its outbound trip drops the last trailer at the first milk-collecting station, then continues on its route for another 15 or 20 miles, uncouples the other trailer and proceeds to the collecting station at the extreme end of the route, where it takes on its load of milk and, on the return trip, picks up the two loaded trailers.

The map reproduced in Fig. 9 shows how unreasonable restriction of the length of vehicles will seriously affect the dairy farmer. If the Kentucky law prohibiting trailers were applied in the State of Michigan, the cost to transport milk by truck into Detroit from a distance greater than 45 miles would be prohibitive, whereas at present milk produced on dairy farms located as far as 125 miles from Detroit is being transported economically by these truck and trailer combinations.

The development of milk transportation in Michigan may be an exception to the rule, but to cover such exceptions the wording in a recommended uniform vehicle code should read, "not more than three units" instead of "not more than two units."

Highway congestion is affected by length restrictions. One truck with two trailers carrying 30 tons of payload, as allowed by the Michigan law, would occupy the maximum highway length of 65 ft., as shown in Fig. 10. Fifteen 2-ton trucks would be required to carry this load; and if each truck were 20 ft. long and the necessary distance of 200 ft. intervened between each two vehicles, these 15 trucks would occupy a total length of 3100 ft. on the highway.

Safe High Speed with Commercial Vehicles

We are entirely in accord with the Bureau of Public Roads recommendation on legal speed limits and believe that a complete revision of truck-speed laws in most States is highly desirable, as the majority of laws now in effect for the purpose of controlling the speed of commercial vehicles were formulated in the days of solid tires and gravel roads.

With the faster speed of modern commercial vehicles made possible by six-cylinder engines and balloon tires, the question of stopping ability is highly important and

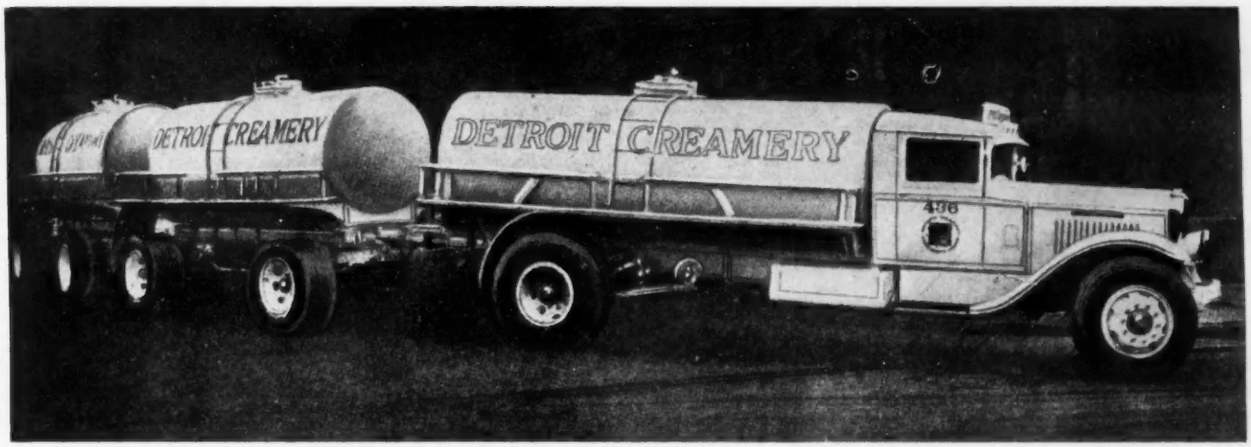

FIG. 8—THREE-UNIT COMBINATION OF TYPE USED IN HAULING 2,000,000 LB. OF MILK PER DAY INTO DETROIT FROM FARMS AS DISTANT AS 125 MILES

power brakes are now standard equipment on nearly all heavy-duty vehicles. Unless the speed-regulation phase of legal activities is carefully watched, we may have to contend with impractical regulations such as have been contemplated in at least one State; namely, prescribing by law a minimum number of square inches of braking surface per 1000 lb. of vehicle gross weight. Such regulations would stifle research and progress in the design of brakes.

From the safety standpoint, regulations now in effect in the State of Connecticut, where safe stopping distances for motor-vehicles of various types traveling at

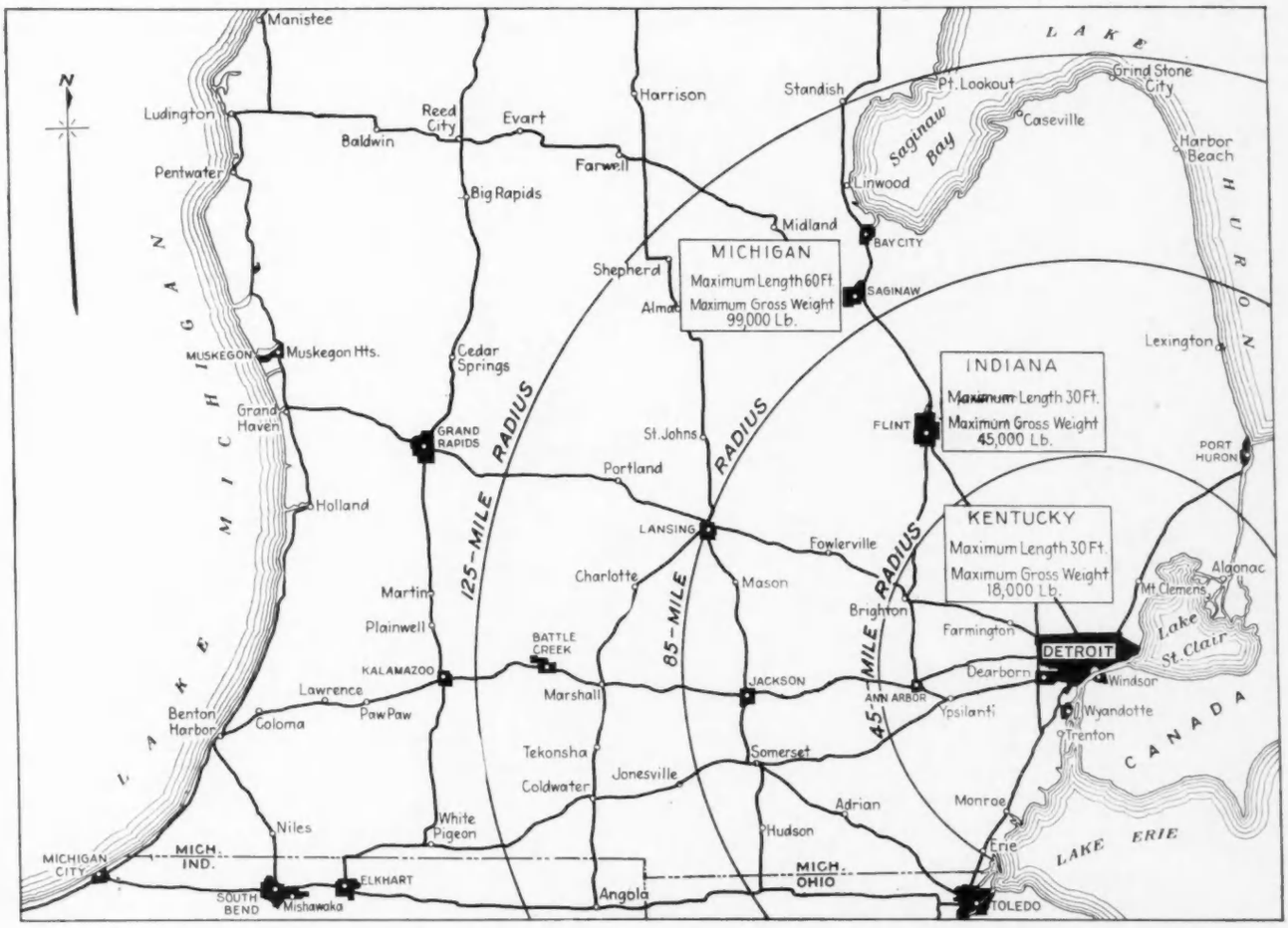

FIG. 9—COMPARISON OF MICHIGAN MILK-PRODUCING AREAS WITHIN PROFITABLE TRUCKING DISTANCE FROM DETROIT UNDER VARIOUS STATE LAWS

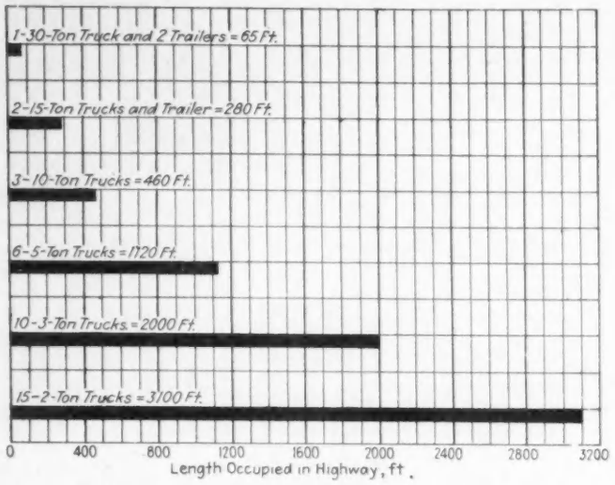

FIG. 10—COMPARISON OF HIGHWAY TRAFFIC CONGESTION BY TRUCK AND TRAILER COMBINATIONS AND SINGLE TRUCKS REQUIRED TO TRANSPORT 30 TONS OF PAYLOAD

various speeds are required by law are very practical. These stopping distances are given in Table 3. Under this regulation, vehicles must have braking systems in good working order at all times and uniform speed regulations must necessarily be based on the deceleration ability of vehicles. Sixteen States now require trailers to be equipped with power brakes. A safe assumption is that, when the 44 legislatures are adjourned in 1933, similar legal regulations will be in effect in almost all of the States.

The safety of commercial vehicles on the highway can best be estimated by comparing accident records as reported by the National Safety Council for the last 4½ years. Although passenger-car fatalities increased in this period by 59 per cent, the safety record of trained truck and motorcoach drivers shows a decrease of 31 per cent for commercial cars.

Complications of Gross-Weight Regulations

A tabulation of four-wheel-vehicle permissible gross weights and single-axle gross-weight limitations presents an extremely complicated picture. Gross weights as shown in Fig. 11 are based upon the States' interpretations of their laws and on the physical limitations of vehicle design and tire capacity. Maximum gross vehicle weights as shown for the four-wheel truck have been computed on the basis of the front-axle weight being not in excess of one-half of the rear-axle weight, resulting in a one-third and two-thirds gross-weight distribution. For instance, while New York State allows a gross weight of 36,000 lb. for a four-wheel truck, the maximum axle weight is 22,400 lb. for the rear axle and one-half of the axle-limitation maximum figures, or 11,200 lb., for the front axle, or a practical maximum gross weight of 33,600 lb.

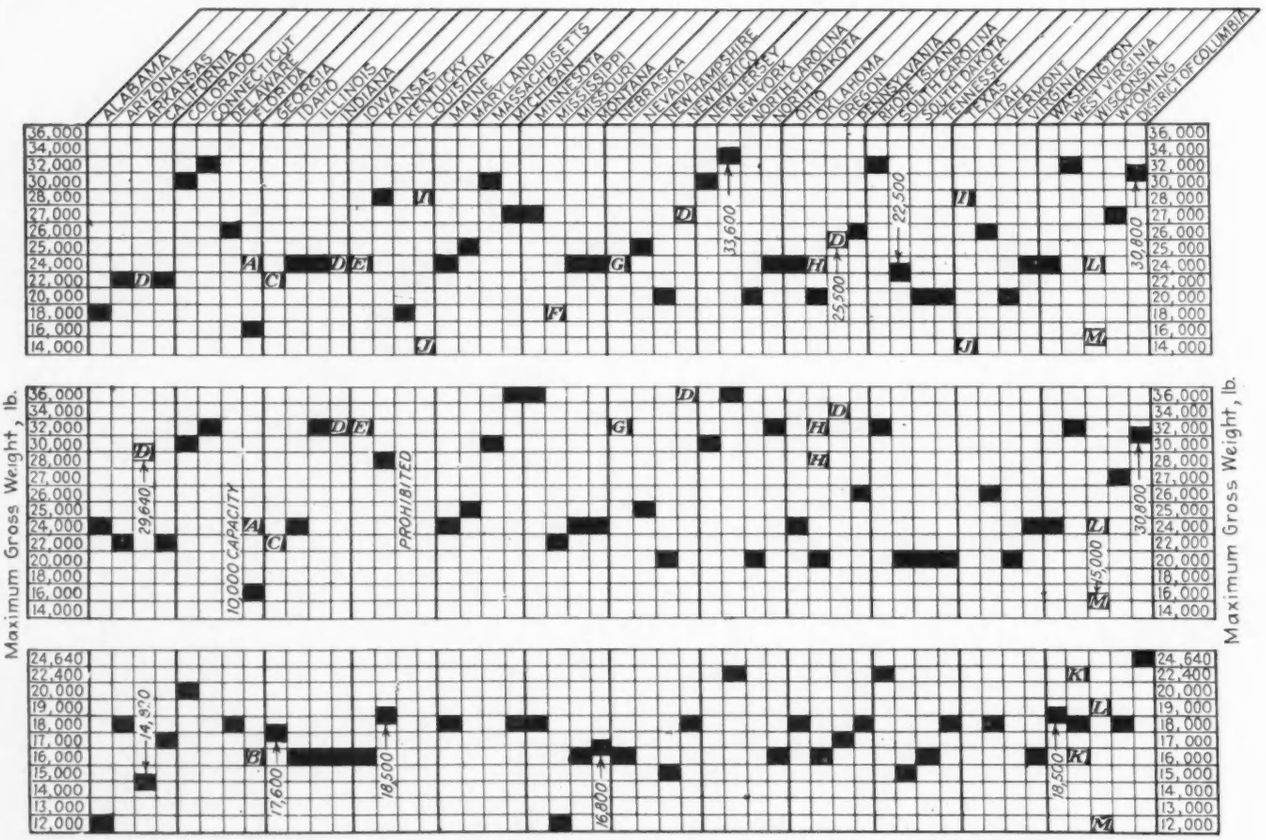

FIG. 11—INTERPRETATION OF MAXIMUM GROSS-WEIGHT RESTRICTIONS APPLYING TO FOUR-WHEEL TRUCKS AND SINGLE AXLE, AS OF OCT. 1, 1932

A—If Operated under Certificate. B—In Counties Specified by Commissioner. C—12,500-Lb. Net Load When for Hire. D—600 ($L + 40$). E—450 ($L + 53\frac{1}{2}$). F—14,000-Lb. Net Load. G—14,000-Lb. Net Load; 16,000-Lb. Farm and Municipal. H—By Special Permit and Fee. I—14,000-Lb. Net Load under Certain Conditions. J—7000-Lb. Net Load. K—Under Certain Conditions. L—On Class-A Highways. M—On Class-B Highways

The almost universal adoption of the pneumatic tire and the impractical legal weight regulations have created problems for the operators which perhaps have not received sufficient attention from designing engineers and no consideration whatsoever from the legislators. Uniformity of tire equipment on front and rear tires on a four-wheel truck and the carrying of the maximum payload permissible by law are demanded by operators. With dual tires on the rear wheels and single tires on the front wheels, it is obvious that with tires of uniform size a weight distribution of two-thirds on the rear axle and one-third on the front axle results in the most practical tire equipment and most economical weight distribution from the operator's standpoint.

Basic legal gross weights for four-wheel trucks range from 33,600 lb. in New York State to 16,000 lb. in Florida. The Texas and Louisiana 7000-lb-payload law brings the gross weight to 14,000 lb. and even less, depending upon chassis and body-weight variations. The nearest we can come to any semblance of uniformity in weight distribution and observe legal restrictions is by grouping those States having somewhat similar gross vehicle and axle-weight regulations, as in Table 4. By such a grouping the State restrictions fall into six distinct capacity groups, and each of these groups has been tabulated to show the maximum gross vehicle weight of a four-wheel truck with the rear-axle load two-thirds and the front-axle load one-third of the total gross weight.

In some cases the result of this interpretation shows a lower gross vehicle weight than that allowed by law. In those cases the maximum permissible axle load is the controlling factor, and the gross vehicle weight has been arrived at by allowing a front-axle weight equivalent to only one-half of the maximum legal weight applicable to the rear axle. The front-axle weights as shown in Group 1 may seem excessive for manual steering control, yet this problem has been solved on our large motorcoaches, the front-axle weights of which frequently range up to 12,000 lb. In this group of seven States, the average front-axle weight is 11,260 lb., the average rear-axle weight 20,800 lb. and the average gross vehicle weight 31,200 lb.

Group 2 shows 11 States having fairly uniform regulations. However, these are not neighboring States but are grouped according to a similarity of vehicles required to best meet existing weight regulations. In these 11 States, the average front-axle weight allowed is 8765 lb.; the average rear-axle weight, 17,535 lb.; and the average gross vehicle weight, 26,300 lb.

Group 3 includes 15 States in which either the 24,000-lb. gross-weight limit or the 16,000-lb. axle-weight limit is the dominant factor. Additional States may come within this classification in 1933, as the legislative trend is toward this class of vehicle. Interpretation shows that weight distribution is uniform in these 15 States: 8000 lb. on the front axle and 16,000 lb. on the rear axle. Louisiana and Texas, with a 14,000-lb. net-weight law, also can be considered to belong in this classification for those trucks serving railroad freight stations, depending, however, upon variations in chassis and body weights.

Group 4 includes only five States with 7382-lb. average front-axle weight and 14,674 lb. on the rear. The difference between maximum and minimum front-axle weights is only 167 lb., and between rear-axle weights, 333 lb. The difference in vehicle gross weight is 500 lb.

Group No. 5 includes nine States with gross vehicle weights ranging from 18,000 to 20,000 lb., with a maximum variation of 666 lb. in front-axle weight and 1334 lb. in rear-axle weight. The average front-axle weight is 6444 lb. and the average rear-axle weight is 12,889 lb., while the average gross vehicle weight is 19,333 lb.

In Group 6 we have only four States. Legal restrictions in Florida allow 16,000 lb. gross for privately owned vehicles. Wisconsin, on Class-B highways, allows 15,000 lb. gross weight. Texas and Louisiana limit trucks passing more than one railroad station to a 7000-lb. payload. In these two States the gross weight of the vehicle will vary according to the weight of the body; therefore we have assumed a chassis and body weight of 6500 lb. plus the 7000-lb. legal payload, or a total gross weight of 13,500 lb.

These problems seem very important to designing engineers and operators, but to the humorist the solution is simple. A sliding weight over the front end and a

TABLE 4—GROUPS OF STATES HAVING SOMEWHAT SIMILAR GROSS VEHICLE AND AXLE-WEIGHT REGULATIONS

State	Legal Weight Limit, Lb. Maximum Gross	Legal Weight Limit, Lb. Maximum per Axle	Weight Distribution with One-Third on Front and Two-Thirds on Rear Axle, Lb. Front	Rear	Total
		Group 1			
New York	36,000	22,400	11,200	22,400	33,600
Rhode Island	32,000	22,400	10,666	21,334	32,000
Connecticut	32,000	.....	10,666	21,334	32,000
District of Columbia	30,800	24,640	10,266	20,534	30,800
Colorado	30,000	20,000	10,000	20,000	30,000
Massachusetts	30,000	.....	10,000	20,000	30,000
New Jersey	30,000	.....	10,000	20,000	30,000
Axle and Gross-Weight Averages			11,260	20,800	31,200
		Group 2			
Kansas	28,000	18,500	9,250	18,500	27,750
Michigan	.....	18,000	9,000	18,000	27,000
Minnesota	36,000	18,000	9,000	18,000	27,000
New Mexico	36,000	18,000	9,000	18,000	27,000
Wyoming	30,000	18,000	9,000	18,000	27,000
Delaware	26,000	18,000	8,666	17,334	26,000
Pennsylvania	26,000	18,000	8,666	17,334	26,000
Utah	26,000	18,000	8,666	17,334	26,000
Oregon	600 ($L \div 40$)	17,000	8,500	17,000	25,500
Maryland	25,000	.....	8,333	16,667	25,000
Nevada	25,000	.....	8,333	16,667	25,000
Axle and Gross-Weight Averages			8,765	17,535	26,300
		Group 3			
Florida (Carrier)	24,000	.....	8,000	16,000	24,000
Idaho	24,000	16,000	8,000	16,000	24,000
Illinois	24,000	16,000	8,000	16,000	24,000
Indiana	24,000	16,000	8,000	16,000	24,000
Iowa	24,000	16,000	8,000	16,000	24,000
Maine	24,000	18,000	8,000	16,000	24,000
Missouri	24,000	16,000	8,000	16,000	24,000
Montana	24,000	16,800	8,000	16,000	24,000
Nebraska	.....	16,000	8,000	16,000	24,000
North Dakota	32,000	16,000	8,000	16,000	24,000
Ohio	24,000	18,000	8,000	16,000	24,000
Virginia	24,000	16,000	8,000	16,000	24,000
Washington	24,000	18,500	8,000	16,000	24,000
West Virginia	32,000	16,000	8,000	16,000	24,000
Wisconsin	24,000	19,000	8,000	16,000	24,000
Louisiana	14,000 lb. payload for trucks serving railroads				
Texas	14,000 lb. payload for trucks serving railroads				
Axle and Gross-Weight Averages			8,000	16,000	24,000
		Group 4			
South Carolina	25,000	15,000	7,500	15,000	22,500
Arkansas	.....	14,820	7,410	14,820	22,230
Arizona	22,000	18,000	7,333	14,667	22,000
California	22,000	17,000	7,333	14,667	22,000
Georgia	22,000	17,600	7,333	14,667	22,000
Axle and Gross-Weight Averages			7,382	14,764	22,106
		Group 5			
New Hampshire	20,000	15,000	6,666	13,334	20,000
Oklahoma	20,000	16,000	6,666	13,334	20,000
South Dakota	20,000	16,000	6,666	13,334	20,000
Tennessee	20,000	18,000	6,666	13,334	20,000
Vermont	20,000	.....	6,666	13,334	20,000
North Carolina	20,000	.....	6,666	13,334	20,000
Alabama	24,000	12,000	6,000	12,000	18,000
Kentucky	18,000	.....	6,000	12,000	18,000
Mississippi	24,000	12,000	6,000	12,000	18,000
Axle and Gross-Weight Averages			6,444	12,889	19,333
		Group 6			
Florida (Private)	16,000	.....	5,333	10,667	16,000
Wisconsin (Class-B Highways)	15,000	12,000	5,000	10,000	15,000
Texas	7,000 lb. payload		4,500	9,000	13,500
Louisiana	7,000 lb. payload		4,500	9,000	13,000
Axle and Gross-Weight Averages			4,833	9,667	14,500

balloon to reduce the weight at the rear have been suggested as standard equipment for the heavier trucks, so that axle loads can be adjusted according to law when crossing our State lines. Seriously, the problem of carrying the maximum payload on a four-wheel truck on pneumatic tires within legal weight regulations can be solved only by shifting more of the payload toward the front. This results in the same type of design as that required for obtaining adequate body load-space with the Indiana and Illinois length regulations of 40 ft. for a four-wheel truck and a four-wheel trailer combination.

So long as solid tires were standard equipment on motor-trucks and many of the States allowed 22,400 lb. on the rear axle, a gross-weight distribution of 80 per cent on rear and 20 per cent at the front was entirely practical. However, balloon tires and restrictive size and weight regulations demand the elimination of waste load space and a more economical load distribution. With tires of uniform size, single front and dual rear, the logical answer for carrying the maximum payload on the most economical tire equipment is a one-third and two-thirds load-distribution factor incorporated in the design of the vehicle.

Digressing for a moment from the main subject, a few facts on how vehicular travel affects highway construction may be of interest. Bulletin No. 28 recently issued by the American Road Builders' Association contains a report covering a most exhaustive study of 6700 miles of concrete pavement in 11 different States. Particular attention was given in this study to cracking of concrete-pavement slabs, for the reason that road builders regard such cracking as primary evidence of weakness or deterioration. The report states:

> The summary of such studies did not show any definite or consistent influence of traffic on transverse cracking where the cross-sections used were adequate to carry the loads imposed on them.

In close proximity to our factory in Pontiac, Mich., we had occasion to make a survey of the effects of vehicular traffic on concrete pavements on United States Highway No. 10 between Detroit and Pontiac. Part of this highway was constructed more than seven years ago and, because of right-of-way litigation, a 500-ft. section of concrete pavement in this highway has never been subjected to vehicular traffic of any kind, yet this particular untraveled section shows an extremely high percentage in lineal feet of transverse cracks, as com-

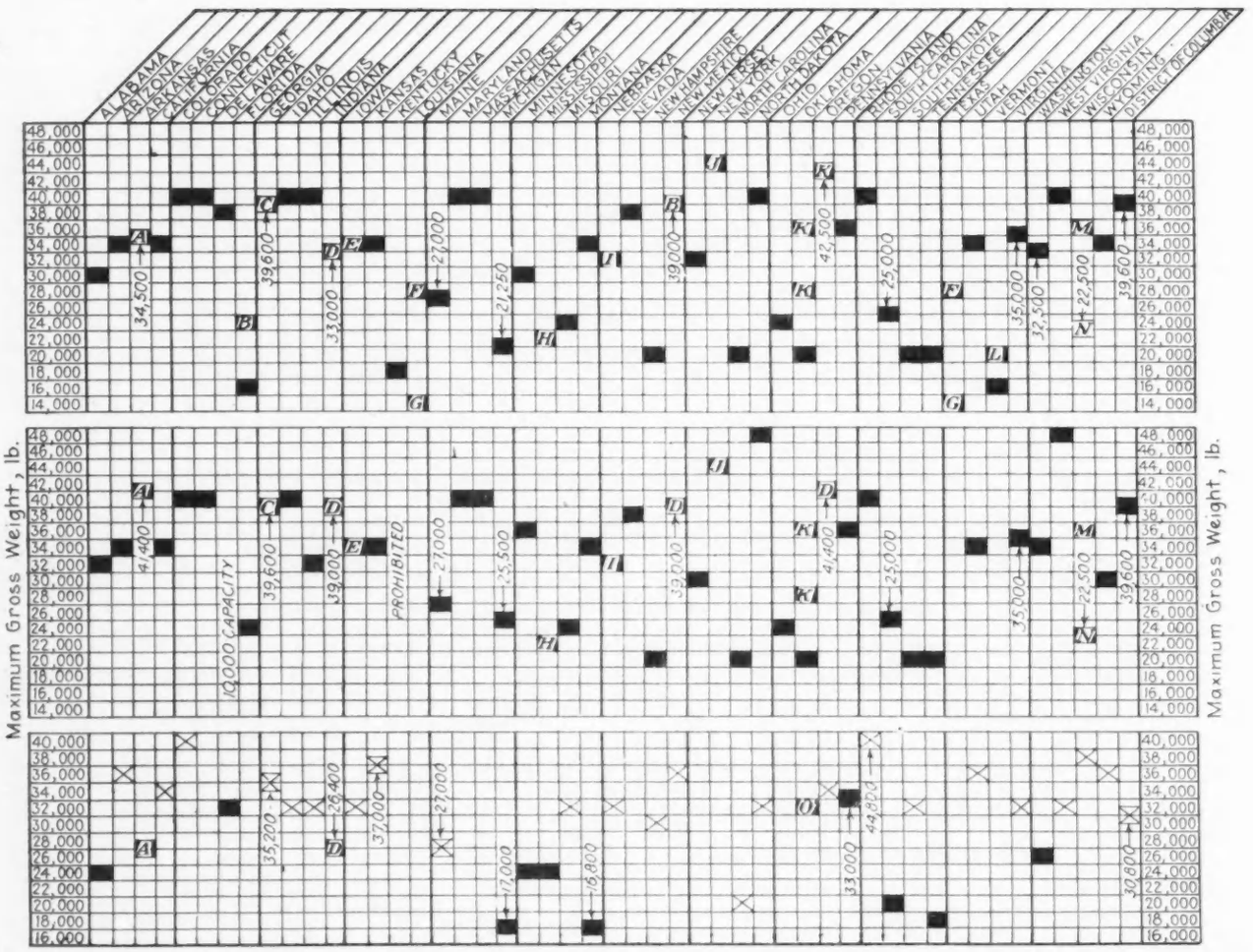

FIG. 12—INTERPRETATION OF GROSS-WEIGHT RESTRICTIONS APPLYING TO SIX-WHEEL VEHICLE AND TANDEM AXLE, AS OF OCT. 1, 1932

A—25,000 Lb. + 650 D. B—If Operated under Certificate. C—12,500-Lb. Net Load When for Hire. D—600 (L + 40). E—450 (L + 53½). F—14,000-Lb. Net Load under Certain Conditions. G—7000-Lb. Net Load. H—14,000-Lb. Net Load. I—14,000-Lb. Net Load; 16,000-Lb. Farm and Municipal. J—30,000 + 750 L; Maximum Gross, 44,000-Lb. K—By Special Permit and Fee. L—On State Aid Roads. M—On Class-A Highways. N—On Class-B Highways. O—26,000 Lb. if Axle Centers Are 46 In. Apart or Less

pared with the remaining part of this highway bearing the heaviest kind of traffic during these seven years.

Our Michigan law allows 18,000 lb. per axle on solid tires. Truck-trailer trains weighing up to 100,000 lb. gross are traveling this highway day and night, transporting heavy tonnages of castings, forgings and other materials to the manufacturing plants in this highly industrialized area. Regardless of these heavy vehicle weights, a survey of a 1500-ft. section shows an average of only 38 lineal ft. of transverse cracks in the concrete, while the 500-ft. section never subjected to vehicular travel in seven years shows an average of 67 lineal ft. of cracks per 100 ft. of pavement.

Four-Wheel Trailer and Single-Axle Weights

Very little consideration has been given by our lawmakers to the basic difference in design between a four-wheel truck and a four-wheel trailer. With a drawbar steering device on a four-wheel trailer and a 50-50 load distribution, the standard practice is to mount dual tires on the front wheels of a four-wheel trailer. A typical example of inconsistencies in legal regulation affecting payload capacity is found in two important neighboring States in which axle-weight regulations are uniform, yet the gross-vehicle-weight limitation is forcing interstate operators to reduce their payload by 12,000 lb. when crossing the State line from Michigan into Ohio. How these limitations affect the four-wheel trailer is evident from the following:

	Maximum Weight Specified		
	Vehicle Gross, Lb.	Per Axle, Lb.	Legal Gross Weight, Lb.
Ohio	24,000	18,000	24,000
Michigan	None	18,000	36,000

Variations in trailer-weight regulations are extremely burdensome for interstate operators but, according to legislative records of 1932, the anti-truck propaganda includes complete elimination of trailers through laws similar to that passed in Kentucky this year.

Legislation controlling six-wheel-vehicle weights is far behind the progress made in the design of six-wheel trucks. A limited number of States have recognized the value of the six-wheel vehicle, but the majority have not yet taken any action to legalize a higher gross weight on six-wheel trucks than that permitted on a four-wheel vehicle. The present situation is highly complicated for interstate operation, as shown by Fig. 12, because of a variety of interpretations on the spacing of axles, gross vehicle weights, axle loads and the $L+$ formula, L symbolizing the distance from the first to the last axle.

Conclusion

Vehicle manufacturers and operators are forced to fit their transportation tools to the legal regulations. Our industry is threatened with even greater complications, and we must be prepared to meet, not only attempts to place additional burdens upon it through higher license taxes, public-carrier permit fees, ton-mile taxes, gross-revenue taxes, higher gasoline taxes, county and municipal privilege taxes and personal property taxes, but proposals at the 1933 legislative sessions for most stringent further restrictions on size and weight. These will affect not only the design but also the earning capacity of commercial vehicles. In some States, notably Texas and Kentucky, operators had to face losses amounting to millions of dollars worth of trucks, the operation of which had to be discontinued.

We are in full accord with the views of the Bureau of Public Roads and the various national organizations that a uniform motor-vehicle code should be recommended to our State Legislatures to stabilize design and operating practice. Unless uniform regulations are adopted, the enforcement of our 48 State laws will require an innumerable force of police officers, which will increase present burdens of taxation and divert a large share of the motor taxes for enforcement of impractical laws rather than conserve these funds for the construction and maintenance of highways, which was the primary reason for their imposition.

Powerplant Needs of Trucks and Motorcoaches

THE powerplant is always a conspicuous and important consideration in automotive development. While notable improvement has been made in the ability factor of trucks and motorcoaches in recent years, we are still far behind the passenger-car. True, it is undesirable to provide for maximum speeds on a comparative basis, but it is essential to provide for increased average speeds. This can be accomplished in part by differences in axle-gear ratios, but this still leaves a gap that can be filled only by increasing the cubic inches of displacement in the engine.

Increasing the displacement obviously introduces a large number of difficult problems. If a high ability-factor is provided for acceleration and hill-climbing, the engine will be operating a considerable part of the time at a low and uneconomical load-factor. From this angle, the constant-pressure type of engine offers attractive features, if it is possible to approximate such a cycle in high-speed and variable-speed engines. As matters stand at present, I am of the opinion that this solution through the present developments of Diesel engines is not very promising. This opinion is based upon such items as weight, cost, smoothness and low specific power output.

The question arises as to how great a displacement per cylinder can be used satisfactorily. Engines are being built today with displacements approaching 150 cu. in., but I believe that it is safe to say that 125 cu. in. is a desirable maximum and 100 cu. in. is far better. Many of the factors which prompt this statement are well known, but one has not been so generally stated. I refer to the engine speed, more particularly the speed produced when the engine is used as a brake. In a passenger-car, the maximum engine speed occurs in driving with wide-open throttle, and the inertia effects of the reciprocating elements are counterbalanced by the pressures developed on the piston. In the truck and motorcoach, the limitations of power and the provision of a governor inhibit these high driving speeds. The limitations of brakes make it necessary, however, to use the engine as a brake with the lower reductions of the transmission engaged, and this frequently causes excessive engine speeds under most unfavorable conditions. This gives rise to the conjecture as to whether it would not be possible to arrange to turn the engine into a compressor and thereby provide a most effective brake, with adequate means for dissipating the heat and at the same time eliminate the destructive results of the present system.

It may be interesting to set up some ideals that seem as remote as some of the outposts of the universe which our astronomer friends talk about. What we want is an engine in which the mechanical motions are all rotary; in which gases at relatively high temperatures and pressure do not come into contact with parts of the mechanism that cannot be provided with direct means of eliminating heat; in which the operating temperature can be maintained at the most favorable point throughout the operating range of speed and load and which will have the same mechanical and thermal efficiency over this range. It should also be possible to exert full torque over the speed range from zero to the maximum. The power from the engine should be delivered to all of the wheels, in direct relation to the traction they have, without the use of mechanical universal-joints or differentials. The body and chassis should be incorporated in one structure so that advantage could be taken of the stiffness it is possible to develop in a structure of this form. The wheels should be independently sprung to the structure by a system that would automatically adjust itself to all conditions of load, speed and road.—From a Metropolitan Section paper by B. B. Bachman, M.S.A.E., vice-president, engineering division, Autocar Co., Ardmore, Pa.

Control of Motor-Vehicle Transport Operations

By F. I. Hardy[1]

Transportation Meeting Paper

THE subject of this paper, while being treated in its broadest sense, will be confined generally to motor-truck operations, the general principles of which, as laid down, will very largely apply to motor-coach operation as well.

Considering the subject in part, control of motor-vehicles means that first consideration must be given to the vehicle. Selection of the right vehicle for the particular service to be performed is of the utmost importance, for, if we do not have a vehicle that is suitable to the work to be done, the best control cannot make a success of management. Many factors must be considered in this selection. A few of them are: the goods to be moved, whether a specific commodity or general merchandise; their weight and bulk; volume of movement per trip, per day and per year; route grades; loading and unloading facilities and town, city, State and Federal regulations as to vehicle size and weight.

Th motor-truck manufacturer is, or should be, in a better position to advise operators as to the best type, size, weight and horsepower of motor-vehicles than the individual operator is in to make his own selection. Fully realizing the competitive situation and that the business of the manufacturer is to sell vehicles, my belief is that, in trying to sell their products, the manufacturers have not made sufficient effort to inform the buyer and advise him as to the type of equipment he should purchase. Even though competitive conditions may ultimately force the manufacturer to sell the wrong type of equipment to the purchaser, he should at least advise the purchaser as to the type that is suitable for the work to be performed, thereby placing himself in a much better position with the purchaser when the unsuitable equipment purchased fails to give satisfactory performance. By doing so, the manufacturer would be aiding in a general movement to further the education of those performing actual motor-transport operation.

Many manufacturers have violated most of the economic fundamentals in selling, as well as in manufacturing, equipment, forgetting the prime fundamental of economical operation, which is that the percentage of weight of equipment to payload must be kept as low as possible.

Mechanical Efficiency Revealed by Failures

Assuming that the right vehicle has been selected for the particular work to be performed, consideration must next be given to its mechanical maintenance and operation. Prices paid for gasoline, oil, tires and repair parts will be found to vary but little among the larger operators, but the cost per mile of operation does vary greatly on these items. This is because of the differences in the system of mechanical upkeep. Good mechanical upkeep will lower the cost per mile for depreciation, tires, fuel and oil; and trucks will also give added revenue and more reliable service, as they will be able to work more miles per day and more days per year. I have found a variation of as high as 20 per cent in tire miles and cost of gasoline and oil, even though the operators were paying the same amounts for these items, because one fleet was kept in excellent mechanical condition while another was poorly maintained.

Most fleet operators have gone to one extreme or the other as to detail in a system for controlling or measuring mechanical condition and operation. The system used should be simple, and the real measure of mechanical efficiency should be the failures per thousand miles, kept in such a way that the operator can visualize an entire fleet. If this is not done, visualizing or controlling the mechanical upkeep is impossible without spending considerable time in making and following up the mechanical records.

My purpose is not to go into details as to the best form of records, but only to state in a general way the essential information that records must show.

Daily maintenance includes examination of tire pressure, gasoline, oil and water, using a simple form that will place the responsibility on the individual if the servicing work is not properly performed.

A mechanical inspection should be made on a mileage basis dependent upon type of equipment, type of work and the like but generally after each 1000 miles. The record covering this inspection also should be simple and enable the responsibility for proper performance of the work to be placed on the individual.

The third and final operation is a complete overhaul of the equipment at some chosen period up to 50,000 miles of operation, this interval being entirely dependent upon the amount of work performed by the equipment. I have generally had this work performed at the end of 35,000 miles. The work done on the vehicle makes it virtually a new truck so far as performance during the next mileage period is concerned; that is, when the truck has a complete overhaul, any wearing parts that will not give good service for an additional mileage period, which we will assume to be 35,000 miles, are discarded and replaced with new parts. Again the type of record used for this general overhaul is such as to place on the individual the responsibility for doing the work right.

Two forms are used for showing failures. One shows the ordinary defects that are found between 1000-mile inspections; the other shows the complete mechanical or road failures. These complete failures, as well as the records of 1000-mile inspections and general overhauls, are entered daily on large fleet charts, each sheet showing a record of 25 trucks for one month. In this way the operator can have in front of him on a few charts the record of a fleet of several hundred motor-trucks showing minor failures and major defects, classified as to type and kind of defect. This enables the operator not only to measure at a glance the efficiency of his own mechanical forces but also to have continually before him the measure of efficiency of the various kinds of equipment and to know the fundamental defects of different makes and sizes of vehicle. Records of this kind enable the operator to eliminate failures resulting from faulty equipment by placing upon the manufacturer the responsibility for failures due to defective equipment or faulty engineering. Cooperation to this end between the manufacturer and the operator is absolutely necessary if we are to improve the manufacture and operation of motor-vehicles.

Control of Operation

The third phase of control has to do with the actual operation or movement of the vehicle. The measure of efficiency varies somewhat with the kind of service re-

[1] Industrial engineer, Boston.

quired; for instance, in retail-store delivery work the efficiency is measured by the number of deliveries per mile and per day per truck; in retail coal delivery, by the tons per mile and per day per truck. The simplest form of record giving the operator these statistics should be used.

It is a fact that in 90 per cent of retail-store and retail-coal businesses one or both of these factors are not known by the operator and that the truck manufacturer either has not tried or could not sell the idea of proper control to the operator. In other words, 90 per cent of the operators, whether using trucks in their own business or for hire, actually do not know the real factors of control that make for the most efficient operation.

Until the trucking business becomes more stabilized and gradually is taken over by large companies and made a big business, the truck manufacturer must do the following three things if the motor-truck is to have its rightful place in the transportation field:

(1) Sell the motor-vehicle of the correct size, weight, horsepower and other characteristics for the service required
(2) Show the operator how to maintain that vehicle in good mechanical condition
(3) Show the operator the correct measure of efficiency in such a concise and simple way that it will be followed by the usual operator

Although generalities are neither conclusive nor instructive, I feel that, in considering the subject broadly, I cannot discuss in detail such items as

Variation of profits in direct proportion to the amount of idle time
Cost of the time that a truck stands idle in the shop
Reduction of overhead expense by increasing the number of hours worked
Cost of delays and loading and unloading time
Reduction of seasonal variations
Satisfying of customers by reliability of service
Sound driver management
Test of suitability of equipment

All I can hope to do is to provoke a general discussion of this subject that may lead to the placing of the responsibility for the education of the operators of motor transport where it belongs at this time.

Improvements in Cylinder Iron

IRON for cylinders presents one of the most important of all automotive-material problems. The steady increase in engine speed has caused heavy heat loads in portions of the cylinder adjacent to the exhaust ports and bores. Irons of 10 or even 3 years ago will not meet present demands. The greatest difficulty of all is presented by grain growth, sometimes called creep.

"Creep" is the continuous tendency of certain irons to grow with every hour of operation. It is entirely distinct from the normal expansion of iron at high temperature. Grain growth can begin at 600 deg. fahr. and becomes acute at 800 deg. Critical portions, such as the center of a piston head or the region around the valves, tend to grow but are restrained by the cooler portions around them. As they cannot grow, they take a certain set when heated and then are subject to terrific stress when cooled, often cracking while standing in the garage during the night.

The problem of grain growth can be met partly by a thorough study of the cooling of the block, but special alloy irons having a low rate of growth and an extremely dense structure also are necessary to meet the situation.

A certain iron has a tensile strength of 60,000 lb. per sq. in., a deflection of 0.20 in. in 12 in., a Brinell hardness between 280 and 300, and its rate of grain growth is slow. We have machined large castings having a Brinell hardness as high as 340, and the iron described can be machined at customary production speeds and feeds. This iron shows very high resistance to valve-sinking, and engines made from it can be operated for 30,000 to 50,000 miles without regrinding the valves. It shows very high fatigue value in an endurance test in a rotating-beam machine similar to that of Prof. H. F. Moore[1], described in his paper on The Significance of Common Physical Properties of Structural Materials[2] and illustrated in Fig. 2.

While its endurance limit of 35,000 lb. per sq. in. is considerably below the 48,000 lb. for S.A.E.-1045 steel, it will run from four to five times as long as the steel in the rotating-beam machine before failure. This may be because it is less rigid when distorted in a breakdown test.

With reference to the relative rigidity of iron and steel, the author, in an article entitled Steel versus Cast Iron for Machinery Building[3], set forth the opinions of seven authorities on the increased rigidity of jigs, machine frames and bedplates made of steel compared with those made of cast iron. The point has also been brought up in some of the technical journals that, if a piece of steel is heat-treated to develop a high Brinell hardness, it becomes very stiff and rigid; if it is subject to deflection, it is more likely to produce high localized stresses in extreme outer fibers in the neighborhood of fillets than is a piece of softer steel or a similar section of cast iron, even though the latter specimens are much lower in tensile strength. It is easy to conceive of a piece of steel heat-treated so hard that, when subject to the slightest deflection, it produces very high stresses in the outermost fibers of the steel.

In contrast with the iron described as having a Brinell hardness between 280 and 300, is a certain iron which shows a Brinell hardness of 150 and has very large carbon flakes. This iron will crack easily and is susceptible to sinking of the valves. Its strength is 25,000 lb. per sq. in. and its deflection in 12 in. is 0.12 in. It grows tremendously; the cylinder head of a valve-in-head engine 30 in. long was 5/32 in. longer after 50,000 miles in hard, heavy-duty service than when it was cast, measured cold. Exhaust valves will sink into this iron as though their only object in life was to bury themselves as quickly as possible.—From a Chicago Section paper by J. B. Fisher, M.S.A.E., chief engineer, Waukesha Motor Co.

[1] Research professor of engineering materials, University of Illinois, Urbana, Ill.
[2] See THE JOURNAL, August, 1925, p. 178.
[3] See *American Machinist*, April 4, 1929, p. 548.

Corrections of Wolf Paper

TWO errors in the paper on Six-Wheel Trucks, by Austin M. Wolf, published in the S.A.E. JOURNAL, October, 1932, p. 387, should be corrected as follows:

On p. 391, second column, line 11, "Torque-arms," etc., should read "Torque-rods extend from each housing," etc.

On p. 400, first column, line 1, "The six-wheeler is not perfected" should read "The six-wheeler is not perfect."

Pneumatic Tires—Old and New

By Burgess Darrow[1]

Transportation Meeting Paper

MUCH study has been given by the author to the history and development of the pneumatic tire in preparation for the writing of this paper. He found in THE JOURNAL of the Society a great wealth of detailed information in papers extending back to 1913 and gives a comprehensive and interesting review of the earliest patent, taken out in England in 1845, and later patents on fundamental ideas. Successive improvements are described, progressively increased section size and reduced inflation-pressures are set forth, truck and motorcoach tire developments and trends are discussed at length and new types of tire that have made their advent in the last two years are described and illustrated.

Among these new tires are the airplane and passenger-car airwheel or super-balloon, the lug-type pneumatic for industrial tractors and road-building machines, the startling development of the airwheel or super-balloon for farm tractors and the pneumatic tire for rail-cars.

Finally, the author tells of some interesting and valuable information gained by his company regarding effects of roads and operating conditions, such as heat, sand and tire load, on the life of tires.

ONE of the major factors that make possible the present-day highway transportation is the pneumatic tire. I shall discuss pneumatic tires in a general way and entirely from the mechanical side, although the chemical developments in the rubber business are fully as important as anything I shall mention in the way of mechanical development.

Many papers on the subject of tires have been given before the Society in years past. Looking through THE JOURNAL as far back as 1913, I found that almost every phase had been well covered. I believe that more information on the theory of tire design is contained in THE JOURNAL[2] than is published anywhere else. Therefore it is doubtful that I can tell much that has not already been referred to in one way or another, but since few of you have occasion to read all of the papers that have gone before, perhaps some of the things I shall say will be new to you. My intention is to state some historical facts rather briefly and then to discuss some of the uptodate subjects, such as tires for tractors, the airwheel for both airplanes and passenger-cars, and the pneumatic tire for rail vehicles; and also I should like to comment upon the present-day pneumatic truck-tire problems.

First Inventor's Broad Understanding of Principles

In 1845, an English civil engineer named Robert William Thomson obtained a patent on the pneumatic tire. In reading the Thomson patent for the first time, I was astonished by the inventor's apparent understanding of the pneumatic-tire principle. He mentions a wide base for the tire; describes the several plies of canvas joined with rubber; the non-skid tread; the single-tube tire and the two-part tire, or tube and casing; describes the conventional tube, various styles of compartment tubes and discloses the use of materials such as horse-hair, balls and springs within the tube to supplement the air and prevent bruises. He mentions tires 4 to 5 in. wide for light vehicles and says that trucks would require larger sizes. He also mentions lessened power consumption, easier motion and diminished noise and states that each part of the tire presents a cushion to the road and will envelop such things as stones. He also mentions speed. Anticipating by 87 years what we are thinking of today, Thomson stated that perhaps we may do without springs and remarked that the tire is adapted to run on timber rails. He showed auxiliary guide wheels but also mentioned flanged wheels. The way we patent things now, Mr. Thomson had material for a dozen patents. But nothing came from his patent.

Forty-three years later, in 1888, the pneumatic tire was born again and patented by Dunlop. This became the real beginning of the tire industry. In 1913, when P. W. Litchfield gave a paper[3] on tires before the Society, Mr. Dunlop, then 74 years old, was present and took part in the discussion at great length. It is not often that we have on record a statement by an inventor, such as Mr. Dunlop, 25 years after his invention has become the basis of a big industry. The Dunlop patent on the tire did not hold, because of the earlier Thomson patent, but Mr. Dunlop said,

> I was the first to patent or protect the endless wire. Mr. Welch was before me but did not mention an air tire in his provisional specification.

Other Early Inventors

Several improvement patents were issued from 1888 to 1892. Pearson's book[4] mentions Thomson and the improvement patents much to the effect that Dunlop obtained a patent in 1888 but soon recognized that Thomson anticipated; Welch obtained an English patent on wire beads in 1890; Brown and Stillman obtained a United States patent on wire beads in 1892; Dunlop bought out both Welch and Brown and Stillman; Bartlett obtained an English patent on climber beads in 1890, for which Dunlop paid $968,000; and Jeffrey obtained a United States patent on clincher beads in 1891-1892; Tillinghast invented the single-tube tire and John F. Palmer patented the cord tire in 1892 and 1893.

Mr. Dunlop spoke of the Palmer tire as follows:

> It will be within the recollection of many how the people went mad with excitement at the introduction of the Palmer cycle tire in England in 1893.

[1] M.S.A.E.—Manager, development department, Goodyear Tire & Rubber Co., Akron, Ohio.

[2] See Appendix to paper for partial bibliography of S.A.E. papers on pneumatic tires.

[3] See S.A.E. TRANSACTIONS, 1913, part 2, p. 208.

[4] See Pneumatic Tires, by Henry Clemens Pearson, (1922), published by *India Rubber World*, New York City.

Automobile building really started in 1898, and from then until now wonderful progress has been made in cars and trucks, and equally remarkable development in tires.

Successive Types of Tire

Litchfield, in his 1913 paper, says:

> During five years, from 1900 to 1905, the clincher tire was perfected and standardized. It seemed that it would have no competitor, but the last-named year brought out the invention of two quick-detachable rims, the Dunlop and the Goodyear. This quick-detachable type of rim became, a year or two later, the American standard, and the wire-bead type of tire began to grow steadily in popularity.

This continent, therefore, took to the straight-side tire in 1905, and by 1916 it was the universal type, except on small cars such as the Ford. Europe, on the other hand, stayed with the soft-bead clincher type for about 20 years longer, or until the advent of the balloon tire in 1924 and 1925.

The next major development in tires was the use of cord in place of canvas, that is, heavy square-woven duck, 17¼ oz. per sq. yd., usually 23's—11 ply, built into the tire to the extent of four to eight plies, according to the size of tire, although European manufacturers often employed a lighter-weight fabric. By the term "cord" is meant parallel strings forming the body of the tire but without any interwoven effect. The use of cord was known in 1892, when it was applied rather extensively to bicycle tires in the form of a very light construction, but virtually all tires for gasoline cars for the first 15 years of the automobile industry, or until about 1914, were of the fabric or canvas type. There were four reasons for this: (*a*) cord, being more flexible than canvas, gave more trouble at or near the rim and beads, and no one had worked out a remedy; (*b*) cord was looked upon, generally, as a frail material; (*c*) the difficulty of making repairs and (*d*) the layers or cords were not easy to prepare and build into the tire. Cord tires consume much less power than fabric tires, however, and therefore were used on electric cars for perhaps six or eight years prior to 1914. Today, cord is used exclusively and canvas tires would be out of the question.

Various types of cord have been used, but only two warrant mention. The two-ply tire of heavy cord, about ⅛ in. in diameter, was employed originally in the Goodrich Silvertown tire up to about 1918. The competing type, which is universal today, is a cord of about 1/32-in. diameter but built up in several layers. It was exploited by the Goodyear company for gasoline cars in 1914, after the company's many years of experience with it in electric-vehicle tires prior to 1914.

History of Pneumatic Truck Tires

The demand for pneumatic truck tires, larger than 6-in. section, began to be felt or appreciated by some about 1912 and 1913, before cord fabric became common; consequently, the earliest truck tires were constructed of canvas and, while some were sold in the next year or two, they were hardly satisfactory. The wider use of cord in 1914 threw a different light on the subject, and, beginning about 1915 and 1916, cord truck tires were produced and a whole new field was opened up for the giant tire.

Goodyear was a pioneer in the matter of truck tires, and to prove their value to the skeptics and at the same time develop the tire for hard service, the company inaugurated, in April, 1917, the Akron-to-Boston Wingfoot Highway Express, which it operated for 2½ years, this being, so far as I know, the first pneumatic-tire long-distance hauling line regularly scheduled on this continent. The tire troubles at the start, in this hard service, were almost unbelievable; the trucks themselves were not much if any better, and the roads were abominable. The first 750-mile trip to Boston required 24 days, with the return made in 7 days. Eventually, a regular schedule of 5½ days was maintained for the round trip. Today, almost any truck can make such a 1500-mile trip in 3 days. I mention the Akron-Boston express experience to show how intensive the truck-tire development was in the early days.

For many years a controversy was waged over the merits of dual truck tires and larger single truck tires, but in recent years duals have met with wider acceptance because of lower costs, larger trucks and better roads, on which the dual shows up to the best advantage. Large singles still remain the choice in rough country; for example, in oil-field work.

Improvements in Tire Construction

Early truck tires developed weaknesses as the result of the effect of heat or over-vulcanization of the rubber in and around the cords, such as had never been encountered in passenger-car tires; consequently, compounding ingredients to resist such heat were found or developed and another big step in tire life was thereby accomplished. As the service requirements of truck tires increased, tire engineers were forced to try some things that they were told could not be done, and sometimes they worked. Canvas strips were run in between the cord plies at the beads to safeguard further against bead and side-wall failures. Theoretically, two such different kinds of fabric—cord and canvas—would oppose each other when used together in the flexing part of a tire and cause trouble, but, like many other forbidden things, it worked.

Strips of fabric called breaker strips, placed under the tread to help hold the tread rubber to the body or carcass of the tire, were made for years of a special open-mesh square-woven fabric, but in truck tires of 1924 we substituted cord-fabric layers for the woven layers, spacing the cords a little wider than in the body plies, and we have used such cord ever since.

Cord fabric was for years twisted in the way to give maximum tensile-strength. About 1923, Goodyear looked for, found and patented a twist for cord fabric which greatly improved the durability of the cord in the tires, and this twist was a certain degree higher than that which resulted in maximum tensile-strength. A slight amount of strength in the cord when new was sacrificed but an enormous gain in durability of the cord resulted and the strength remained to the cord after the tire was run. The following table shows the gains that we consider were made by the new cord development over the old, in the case of severely overloaded truck tires of, say, 5-in. size:

Tire-Life Periods	Tensile Strength, Lb.	
	Goodyear Old-Style Cord	Goodyear Super-Twist Cord
New	16.5	15.5
8,000 miles	14.0	15.0
16,000 miles	6.0	13.0
24,000 miles	Too weak to run	11.0

How Passenger-Car Tire Size Has Increased

An important trend in tire development from the very start has been the increase of cross-section size for a given load, usually accompanied by a corresponding decrease in the inner wheel or rim diameter and, obviously, a decrease in the necessary tire pressure, and with the possibility of fewer plies. For example, the Cadillac automobile, which has been a large car almost from the start, with some weight changes up and down from year to year, once ran on tires of actual cross-section not exceeding 4.5 sq. in. and perhaps less, and since then the tire size has been gradually increased,

going through stages of 4.95, 5.50, 5.75, 6.75, 7.00 and now 7.50, while the rim diameter has come down from 27 to 17 in.

Four outstanding steps have been taken in the passenger-car tire-size trend, in addition to this occasional stepping up of the tire size by the car manufacturers while they were within one of the four stages. With the introduction of straight-side tires, Goodyear turned to the principle of 10 per cent more volume than was common practice with clincher tires. With the advent of the cord tire in 1914, the same principle was extended by Goodyear to 10 per cent extra width over the nominal dimension. In 1923-1924 another step in oversizing occurred, resulting in what was called the balloon tire; in effect, about another 20-per-cent increase in width over the tire it replaced. And now we are entering, or have entered, the super-balloon stage of "doughnut" or "airwheel" tires, which are 30 to 40 per cent greater in width than the balloon tire.

Truck Tire-Size Trend

Truck-tire dimensions were naturally affected by the passenger-car results. Starting with little or no oversize, truck tires were soon made 10 per cent oversize and gradually grew until 15 per cent oversize became the accepted figure; then finally, in 1928, came the truck and motorcoach balloon tire, which is really 10 to 15 per cent further oversize and operates in the range of 50 to 70-lb. pressure, as compared with 90 to 110 lb. for the high-pressure type. The question immediately arises, Is there to be a super-balloon stage for truck tires? The answer certainly is not clear to me. In the case of small trucks, practically passenger-car chassis, or in the case of trucks for special uses, the super-balloon, I believe, is surely ahead of us. In the case of larger conventional trucks, three factors must be considered: (*a*) the small-diameter-rim idea can be carried to such an extreme that tire life suffers, (*b*) the brake problem and (*c*), probably most important, is the question of finding room on the chassis for tires any larger than those we now have. This brings up the question of truck-width limitations, which I shall not attempt to discuss.

Many Minor Details in Development

Many other details in the development of pneumatic tires might be mentioned, but I shall merely list a few, such as tubes, puncture-proof and otherwise; valves, including the latest trend toward the convertible type for truck tires; rims; impact tests, largely conducted by the Bureau of Public Roads; increase in mileage year by year; trends in tire pressures and number of fabric plies; studies of power consumption; facts regarding second and even third-grade lines of tire; the confusion of tire sizes; special-brand tires; tests on heating of tires; materials, such as wire, cotton and rubber, from which tires are made; tire specifications, particularly the Government specifications; how endurance or test-car tests are conducted; evolution of tire-building and factory-control methods; industrial facts; tests of non-skid tires; necessity of venting or providing an outlet for trapped air between tube and casing, in the case of certain rims, particularly the drop-center type; tire noise, in the case of passenger-car tires, and the remedy; and I might discuss selling on a mileage basis.

The New Airplane Airwheel

The full super-balloon tire, or what Goodyear terms an airwheel, such as is now widely used on airplanes and which may be, for example, a 9-in. tire on a 3-in. hub or a 15-in. tire on a 6-in. hub, as shown in Fig. 1, seems to me to be one of the most astonishing tire developments going on at this time. It is astonishing because of its extreme proportions, that a way was ever devised to build such a tire and because of what it does on an airplane. It is not to be confused with the tires now coming into use on passenger-cars and bearing the same name, because, while the passenger-car tires are as big in cross-section and as small in rim diameter as the application to and behavior on the car will permit, the passenger-car airwheel is far from being like the airplane tire in proportion of tire cross-section to rim or hub diameter.

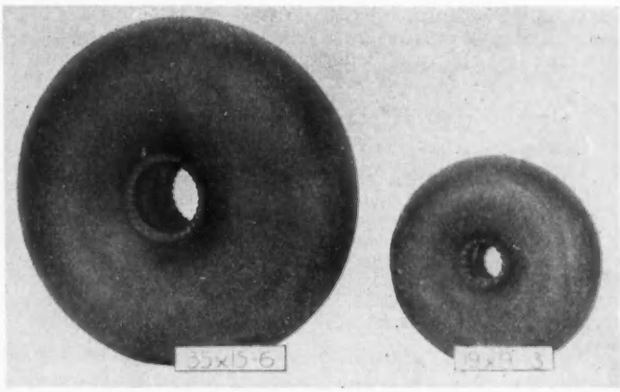

FIG. 1—TWO SIZES OF AIRPLANE AIRWHEELS, FORERUNNERS OF THE AUTOMOBILE AIRWHEEL

The high-pressure tires formerly used on airplanes were high-wheeled affairs of relatively small cross-section and operating at 50 to 60-lb. pressure. The airwheel operates at 12½ to 20-lb. inflation. In the small and medium sizes the airwheels have a slight weight advantage, but this turns to a slight disadvantage in the very large sizes for large airplanes. Airwheels taxi through and take off and land in mud where the old-style tire cannot operate. The ability of this "all-balloon" tire to yield sidewise and absorb shocks, both vertical and lateral, makes landings and ground maneuvers possible with safety that, if duplicated on tires of the older type, would result in failure of the landing-gear or cause the airplane to ground-loop or nose over. The safety of the airwheel is its most important feature, especially for experienced pilots when making difficult or emergency landings and also for beginners. The new tires offer very little more air resistance than the old style.

Brakes for airplane airwheels presented a problem, with only 3 to 6 in. of hub diameter in which to work, but the brake now available is entirely adequate, smooth-operating and reliable. Fear that it might not be possible to brake the all-balloon tire satisfactorily led some tire companies to bring out for airplanes another range of tire sizes with somewhat larger hubs, sometimes called "semi-balloons", but some of the desirable qualities of the all-balloon have had to be sacrificed in the intermediate tire.

Certain few airplanes have been operated on airwheels with all the usual shock-absorbing struts eliminated, thereby placing the whole burden of the landing shock on the tires. Within reason, this type of operation may be feasible, but no broad claims are being made for it.

Fears that a tire of this sort might prove dangerous if it went flat before landing have proved groundless. My guess is that, although the deflated airwheel causes the airplane to lurch, it gives the pilot a chance to correct for it before the hub strikes the ground and causes trouble, whereas a deflated high-pressure tire gives the

pilot no chance to correct his controls before the airplane is dragged around by the flat tire and the wheel.

The airplane airwheel is making satisfactory progress in the United States but is making greater headway in several European countries, and this no doubt will hasten full acceptance on this continent.

The Passenger-Car Airwheel

The passenger-car airwheel, illustrated in Fig. 2, no doubt is an outgrowth of the airplane tire, but this development would have come eventually independently of the airplane. The hub diameter for passenger-car tires of the airwheel type is fixed by the brake diameter over which the rim fits. In this respect the automobile airwheel, with its 14 to 16-in. rim, is much nearer to the balloon tire it replaces than it is to the airplane tire, which is truly "all tire". It has met with considerable acceptance, and with its strong inherent advantages and the few disadvantages, which I believe will be overcome, we can look forward to its complete acceptance, eventually, on passenger-cars.

The automobile airwheel absorbs road shocks to an almost unbelievable extent. This protects the passengers and the mechanism from the shocks and makes possible much higher speeds over bad roads. The tires, with their greater road contact, provide better traction and braking and will travel successfully over very soggy or soft ground that would stop the older forms of tire.

Plenty of objections to such tires are raised at present, but I do not regard any of them as insurmountable. While the tires cushion all severe jolts and jars, a disagreeable rocking motion is felt which would be very desirable to eliminate. A tendency to induce shimmy and tramp is inherent but no doubt will be remedied. A tendency also is present for the treads to wear fast and in a peculiar cupped manner, owing to the low pressure, but the cupping is largely a matter of appearance late in the life of the tire. Hard steering when parking also is a problem that will be met and solved.

Airwheel tires behave somewhat differently than do others when cutting around traffic, rounding corners and following ruts or car tracks, but this presents no hazard and simply requires a little experience. Whatever tire-noise problems have arisen with balloon tires can be expected to be accentuated in the case of airwheels, with a little more of the non-skid-vibration type of noise and considerably more road-swishing noise and drumming coming from brick pavements and the like.

I believe that the chance of blow-out and punctures from nails is lessened, but the effect of sudden deflation is still a debatable point. A 7.50 airwheel, when suddenly deflated, lets a wheel down about 4 in. which is considerably less than is commonly supposed and is about ¾ to 1 in. more than the drop with the corresponding size 6.00 balloon tire. A considerable number of shot-gun tests have been made and the fact that car engineers seem to be adopting or designing for the airwheel leads to the conclusion that the majority do not consider the new-style tire a serious hazard.

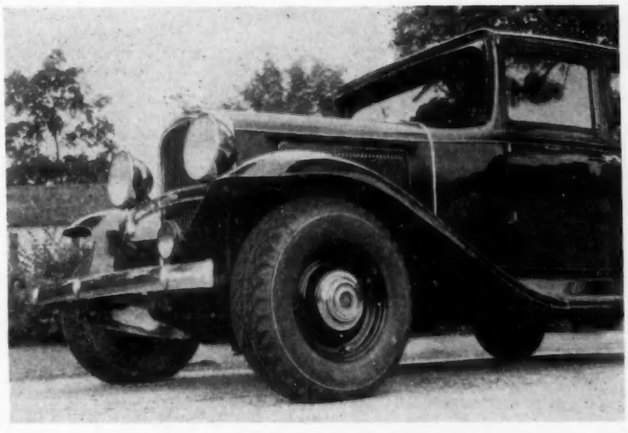

FIG. 2—SUPER-BALLOON AIRWHEELS ON A PASSENGER-CAR

Improved Appearance Is Regarded as a Powerful Factor in Creating the Demand for This New Tire Type

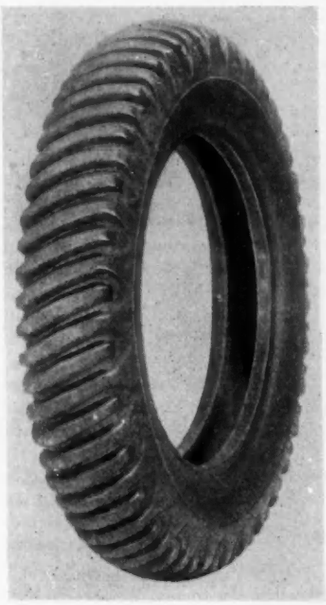

FIG. 3—PNEUMATIC LUG TIRE FOR INDUSTRIAL TRACTORS AND ROAD MACHINERY

The prevailing pressure for the new tires will be about 20 lb. for small cars and up to 24 lb. for the larger cars. Airwheel rims are all of the one-piece drop-center type, but the small rim diameter has made advisable the building of a slight taper on the rim under the tire beads so that the beads can wedge slightly and avoid creeping, such a taper having been employed on airplane rims for years.

Airwheels for cars seem to have been resolved down to rim diameters of 15 and 16 in., with the 17-in. rim a possibility for some big cars, because of the required brake diameter, and the 14-in. rim a possibility if a relatively large airwheel is desired without exceeding a certain over-all tire height. Rims of 15-in. diameter will take up to 13-in. brakes, and 16-in. rims will take up to 14-in. brakes. Apparently a way will have to be found to reduce the brake diameter on the larger cars an inch or more to 14 in. unless 17-in. rims are to be used.

Occasional mention is made of the possibility of eliminating springs if tires are made big and soft enough. However, it seems to me that a tire is fundamentally intended to envelop small road obstacles and absorb jars from irregularities, whereas the springs, aided by shock-absorbers, take care of the big upward thrusts of the tires, which have no snubbing action.

From an appearance standpoint, a moderate-size passenger-car super-balloon tire, on which the industry is focusing its attention, has, in the opinion of most persons with whom I have talked, such an advantage over all former styles that appearance is almost reason enough in itself for adoption.

The following table represents, I believe, the best thought today as to sizes of super-balloon tires for substitution for balloon tires:

Balloon Tires, In.[a]	Super-Balloon Substitute, In.[a]
5.00	6.00
5.25	6.50
5.50	7.00
6.00	7.50
6.50	8.25
7.00	9.00
7.50	9.00

[a] Nominal.

Industrial tractors, farm tractors and road-building machines are forms of vehicle that are undergoing a

FIG. 4—AIRWHEEL-EQUIPPED FARM TRACTOR ON WHICH THE FARMER NOW ENJOYS A LUXURIOUS RIDE

Demand for This Type of Tractor Tire Arose in the Sandy Florida Orange Groves and Has Extended to Other Regions Where Good Traction in Soft or Sandy Soil Is Needed

change with regard to pneumatic tires. The industrial tractor and road-building machines have used solid rubber tires for years and of late have occasionally been equipped with standard pneumatic truck tires; but the time has now come when special tires are warranted for their use. These are being offered under such trade names as Pneumatic Lug tires or Road Builder tires, to distinguish them from standard tires.

The lug-type tire, shown in Fig. 3, matches certain of the prevailing pneumatic truck tires in size, for example 6, 7 and 8 in., but is of lighter construction with lower rated carrying-capacity and with an exaggerated non-skid or traction-tread design. The wheel load on these machines is not excessive and overload in most cases is impossible; the need is for a tire of generous width that will not tend to sink in soft ground; that has a tread giving the utmost traction, as near like cleats as possible, and that is somewhat less expensive than the standard pneumatic truck tires previously sold for such use. Some industrial tractors attain relatively high speeds and speed brings a demand for pneumatics.

The startling development, however, is the use on farm tractors of the full airwheel, practically the same tire as was developed for airplanes except that, of late, a non-skid tread has been added, as shown in Fig. 4. Conversion of a hard-riding, bumpy, steel-wheel tractor into a luxurious conveyance with pneumatic tires 20 in. wide and the cushioning that goes with 3 to 5-lb. inflation-pressure seems to be a bold conception. The demand came first from the orange groves of Florida, where sand hindered the usual operations and steel tires sometimes damaged tree roots. Other demands have sprung up all over the Country, from golf courses and various soft and muddy field operations. The drawbar pull that can be exerted by the tractor with such tires is astonishing; in most tests it exceeds that with the steel tires. Where the steel-lug tire sometimes digs itself in and becomes stuck, the airwheel stays on top of the ground. As the tire carcass was not very thick, it punctured rather easily, but this was overcome by the use of thick rubber liners.

Airwheels in the sizes required for tractors are expensive, and it is likely that, while they no doubt will be required in some classes of service, they will have to yield in volume of sales to some intermediate size of tire that can be built cheaper but will retain the advantages of the full airwheel to a certain extent.

Pneumatic Tires for Rail-Cars

The pneumatic tire for rail-cars will make the use of pneumatic tires on all types of vehicle about complete. Considerable publicity has been given to the Michelin pneumatic-tired railroad car since last fall, and I welcome this opportunity to give some of the tire facts. The idea of pneumatic tires for use on rails is not new; as previously stated, even Thomson, in 1845, mentioned that his tire might be used on "timber rails", but, so far as my knowledge goes, nothing much ever came of any of these thoughts except to put on rails an occasional conventional automobile fitted with steel wheels. The Michelin Tire Co. announced a successful pneumatic-tired railroad coach sometime in 1931, after several years of intensive development during which the French company built nine different units and determined the underlying principles for a successful operation of this kind. Now the Edward G. Budd Mfg. Co. is developing the Michelin idea on this continent, having completed one coach last spring, which is shown in Fig. 5, and is building several at present for various Eastern railroads. The company I represent is at present the only one licensed to manufacture tires of the Michelin design under the Michelin patents.

The rail tire is a 35 x 5.5 size on a 24-in. conventional straight-side rim and is of the high-pressure type, operating at about 100-lb. pressure. The wheel flange is a continuation of the rim extended out around the side of the tire and terminating in a conventional rail-flange shape. A ring of wood or metal inside the tire limits the drop in case of deflation so that the flanged wheel will not interfere with the fish-plates. The inner tube partly encircles the safety ring, making the tube more or less crescent shaped. The tire tread is somewhat flattened and rather wide and has a non-skid pattern designed especially to clean the rails. Michelin has equipped the tires with signals that sound an alarm in

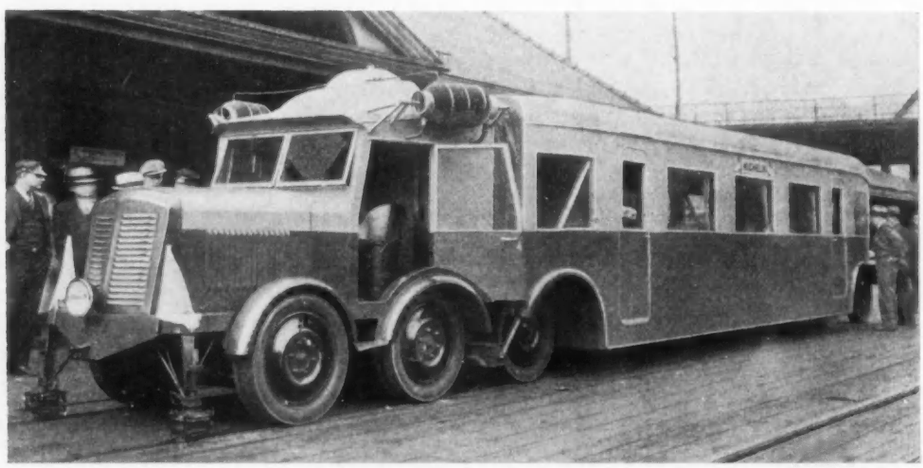

FIG. 5—MICHELIN RAIL-CAR EQUIPPED WITH PNEUMATIC TIRES ON FLANGED RIMS

These Tires Are of High-Pressure Type with Wide Tread and Enclose a Ring of Wood or Metal To Prevent Collapse in Event of Deflation. The Inner Tube Partly Surrounds the Ring in Crescent Shape. This Rail-Car Development Is Believed To Be Destined To Have Great Importance

the driver's compartment in case of an under-inflated tire.

One of the Michelin cars was in the United States for a few months and was demonstrated in and around Philadelphia and in the vicinity of Akron, Ohio. It is about 45 ft. long, has a driver's cab, a small baggage compartment, a compartment for 24 passengers and a large rear baggage compartment. It weighs, empty, about 13,500 lb. and, loaded, about 16,500 lb. At the forward end is a three-axle truck and in the rear a two-axle truck, giving a tire loading of 1650 lb. per tire. It was powered by a small four-cylinder Panhard engine capable of driving the car at a speed of 55 to 57 m.p.h.

Ability of the car to pick up speed and stop quickly means that it can make a high average speed regardless of stops. It is extremely quiet, as no rail or tire noise is produced. The weight of the car per person carried is about one-fifth the weight per passenger of the usual railroad car. This accounts for the low initial cost of the unit and the fact that it can be carried on pneumatic tires.

Our company is following intensely the development of this tire. We know that the load rating can be increased appreciably to enable the tires to support a heavier vehicle without unduly increasing the number of axles. In this development we obtained some interesting figures on tire life with various overloads. The following table is for a tire known to be able to carry a 2000-lb. load for 20,000 miles in rail service, and the figures show the mileage expectancy when the tire is loaded more or less than 2000 lb.

Load, Lb.	Tire Life, Miles
1,800	28,600
2,000	20,000
2,200	11,600
2,400	6,300
2,600	2,800

A 30-per-cent increase over the correct load reduced the tire life to 14 per cent of normal.

One may theorize at length as to the future of this rail-car development for passenger and/or express and freight service, for main lines or branch lines, and single coaches or trains, but it will not be very long before we shall know the possibilities, for cars of this type are in use on three French railroads and this Country will soon be getting experience. Michelin undoubtedly started a new line of activity and is making an important experiment together with the Budd company. The particular type of tire now used may be the final solution, or a better tire may be developed. Having worked on the problem for some months, I am inclined to think that this new form of vehicle may have a very revolutionary effect on rail travel.

Present Truck and Motorcoach Tires

Requirements of the pneumatic truck tire are becoming more severe every year. Faster acceleration and deceleration, higher speeds and longer runs are some of the reasons for expecting lower mileage; yet, the tire mileage increases. Headway has continued to be made, I believe, because, (*a*) the tires are built better, (*b*) the truck and motor-coach manufacturers are appreciating the wisdom of correct tire sizing, (*c*) the operators are more and more demanding adequate tires and (*d*) tires are being cared for better by maintaining inflation pressures and avoiding other abuses.

The matter of tire sizing has been referred to so often, year after year, that I almost hesitate to bring it up again. As pointed out in the rail-tire discussion, a tire properly rated for 2000 lb. gave only 14 per cent of its normal life when overloaded 30 per cent, whereas it gave almost 150 per cent of its normal life when underloaded 10 per cent. These figures would not be true to such a degree for the average truck tire, but in certain truck operations that involve speed and heat they might very likely hold true. The phenomenal mileages on motorcoach tires today is due largely to the fact that the coach builders and operators have stepped up the tire size until the tires are underloaded 10 per cent or more.

Rims also have much to do with tire life, and I am glad that truck manufacturers are paying more attention to this than formerly. When the 9/10 rim (7.33 in. wide) was introduced, it did a world of good to the 9-in. high-pressure and the 10.50 balloon tires by replacing the 8-in. rim (6 in. wide). The same principle applies to all other sizes.

Tire heat is an old subject. From the results obtained with thousands of tires, we estimate that motorcoach tires which give 28,000 miles in a Northern belt, through Ohio, Michigan and Minnesota, will give only 20,000 to 22,000 miles, or a reduction of 25 per cent, when used in the South, solely because of tire-operating temperatures. To state just what tire temperature is dangerous is difficult, but our guess is that heat above 170 deg. fahr. will rapidly reduce the tire life. Some interesting temperatures reported not long ago were 99 deg. official, 113 deg. 5 ft. above the road and 137 deg. at the road. In another test we found a temperature of 239 deg. in the tire itself.

In some of our motorcoach-tire operations we discovered a very interesting fact. New tires were applied in the colder seasons and removed after a certain mileage and held for summer use. We found that the partly used tires stood the hot-weather running better than new tires, no doubt because they were worn thinner and developed less internal heat and, also, possibly because they were broken in and the carcass limbered up before subjecting them to heat. Incidentally, this scheme provided tires with newer and better non-skid treads in the winter.

We have made some comparisons of tread wear in various parts of the Country which are of interest. Obviously, the character of sand available for road building in different localities will affect tire wear, some sand being hard and sharp and other sand not so sharp. We call the tread life at Akron, Ohio, 100 per cent, and find it much shorter in Florida, with a rating of 60 per cent, and much longer in Arizona, with a rating of 160 per cent. Hills and curves also have an important effect on tread wear.

These are some of the conditions that cause tire service to vary so greatly in different parts of the Country and at different seasons. However, unless tire design and manufacture are watched and checked in the most thorough manner, tires of equal quality of raw materials will vary widely among themselves. About 100 miles of cotton yarn is used in a typical pneumatic truck tire. This is twisted into about 8 miles of cords, then assembled in the tire in about 25,000 strings or cords each perhaps 20 in. long. In and around the cords is a coating of rubber, and reinforcing elements are added at the various critical points. It is rather obvious that care must be used to twist and assemble the cords uniformly and to rubberize them without the slightest imperfection; and that considerable skill or experience can be exercised in the way that reinforcements are added. We have seen, from the discussion of heat, that bulk of material is no criterion of tire performance. Given certain raw or semi-raw materials, it is a nice art to be able to build a tire of the highest quality.

The pneumatic truck tire, made primarily from vegetable matter in the form of rubber and cotton and carrying an individual tire load of 2000 to 8000 lb. depending on size, which revolves 10,000,000 times in

delivering its 25,000 miles of service and, in so doing, takes all the road shocks, all the driving power and all the braking, is a tremendous achievement resulting from 40 years of ceaseless effort. And I can say without any hesitancy that the work of improving tires will go on and on.

APPENDIX

PARTIAL BIBLIOGRAPHY OF S.A.E. PAPERS ON PNEUMATIC TIRES

Pneumatic Tires, P. W. Litchfield, TRANSACTIONS, 1913, part 2, p. 208
Size and Inflation of Pneumatic Tires, P. W. Litchfield, TRANSACTIONS, 1915, part 2, p. 194
The Pneumatic Tire and Rim Situation, J. E. Hale, TRANSACTIONS, 1916, part 2, p. 306
Power Losses in Pneumatic Tires, E. H. Lockwood, TRANSACTIONS, 1917, part 1, p. 387
Pneumatic Tires on Trucks, B. B. Bachman, TRANSACTIONS, 1919, part 1, p. 459
Shoeing a Car with Low-Pressure Air, J. E. Hale, TRANSACTIONS, 1923, part 2, p. 501
Public's and the Car Builders' Attitude toward Balloon Tires, J. E. Hale, THE JOURNAL, December, 1923, p. 461.
Adapting Motor-Cars to Low-Pressure Tires, J. E. Hale, TRANSACTIONS, 1924, part 2, p. 322
Pneumatic Tire Elements and Development, L. J. D. Healy, THE JOURNAL, April, 1925, p. 421
Balloon Tires for Use with Drop-Center Rims, B. J. Lemon, THE JOURNAL, June 1926, p. 623
Tire Production Progress, B. J. Lemon, S.A.E. JOURNAL, May, 1928, p. 508
Tires and the Rim Riddle, B. J. Lemon, S.A.E. JOURNAL, March, 1931, p. 332
Judging the Super-Balloon Tire, B. J. Lemon, S.A.E. JOURNAL, October, 1932, p. 403.

THE DISCUSSION

SIDNEY M. CALDWELL[2]:—Mr. Darrow has very ably presented the history of the development of pneumatic tires to meet the ever-increasing severity of transportation service requirements. It seems to me very pertinent now to consider how well modern truck tires are really doing their job and what can be done in the way of further improvement.

Mr. Darrow indicates in his paper that tire life is inversely proportional to the square of the load. Our own data coincide with his as to the economic loss due to overloading. The picture of original tire equipment on trucks and motorcoaches of 1932 is alarming with respect to overloads imposed on tires when only the rated load is considered. We find tire overloads ranging from 11 to 101 per cent; and we all know that double and triple the rated loads are common. The study of motor-truck transportation recently completed by the Department of Commerce shows that one-fourth of all trucks operate at 100 to 125 per cent above their rated loads.

The results of this abuse of tires are clearly shown in the percentages of tire adjustments against sales reported by the Rubber Manufacturers Association, covering virtually all tire manufacturers in the United States, as given in the accompanying table.

Tire Size, In.	Adjustments, Per Cent
5	4.17
6	7.09
7	12.08
8	15.85
9 and 10	21.28
5.50 to 7.00	2.75
7.50	7.38
8.25	8.32
9.00	8.86
9.75	9.83
10.50	17.90

As compared with these truck-tire figures, we find that for private passenger cars, on which overloading is not a factor, the percentages are much lower. For all sizes of passenger-car balloon tires the figure is 1.94 per cent.

In addition to the truck-tire overloads imposed by direct weight of vehicle and load, the problem of further overload on inside duals caused by negative axle camber is presented. We have found several cases in which this condition placed an additional 10-per-cent overload on the inside tire. If the road crown is also considered, this extra load becomes 20 per cent.

Tire engineers and chemists can be depended upon to constantly improve the product, but what we need today is a better and more sympathetic understanding between vehicle, transportation and tire engineers. The best truck tires the industry knows how to produce are not doing a good average job. This is proved by the figures given, which also prove that their failure to do so is not the fault of the tires. What can we all do to improve the situation?

[2] M.S.A.E.—Director, development division, tire department, United States Rubber Co., Detroit.

Bad Conditions Created by State Vehicle Laws

COMPLIANCE with State motor-vehicle regulations is at present almost a physical impossibility, as the State laws were drafted and their passage brought about by interests opposing the commercial use of the highways, without due consideration of industrial demands, engineering principles and valuable data on road impact made available by the Bureau of Public Roads.

The result is at once noticeable on our highways. Light trucks are dangerously overloaded and trucks designed to carry the average loads necessary to serve industry are barred by unreasonable weight restrictions. These heavily overloaded trucks constitute the greatest danger on our highways.

Truck operators in general are willing to comply with the laws of their State and of those States which they enter in the transaction of their business, but, when their equipment enters a foreign State, it may be outlawed and the drivers arrested after they are miles within the State. The fines, long-distance telephone calls and delays cannot be regained out of current earnings, as competition has forced trucking rates down to the lowest possible level. This is due largely to the fact that many truck owners are ignorant of operating costs, as is shown by recent investigations of the Department of Commerce.

One of the most important activities of the motor-trucking interests should be an effort to bring about the elimination of motor tag wars, which build walls around our States, by working for universal reciprocity on State license tags. Tag wars have caused bitter feelings, sometimes leading to costly boycotts.—From Transportation Meeting paper by H. C. Kelting, chairman, Truck Association Executives of America.

Vibration of Instrument-Boards and Airplane Structures

Aeronautic Meeting Paper

By Stephen J. Zand[1]

THIS paper supplements one on the same subject by the same author, published in October, 1931, and describes a new and improved three-component vibrograph with which separate vibrograms of the three components can be obtained simultaneously and with considerable magnification.

A short mathematical analysis is given to show the fundamental difference between vibrographs and accelerometers.

Effects of vibration on different instruments are discussed and the approximate maximum permissible amplitude of vibration at cruising-speed frequency for various instruments is presented in a table.

Consideration is given to correct design of instrument-boards and their suspension in an airplane, and the theory of forced vibration with damping is reviewed to show that, if not rightly chosen, shock-absorbing materials can do as much harm as good.

Other subjects discussed are the physiological aspect of vibration, the sources of vibration, the common peculiarities of vibrations from similar types of installation and miscellaneous uses of the vibrograph.

The author concludes with recommendations as to further study of vibration problems in airplane design.

SINCE the publication of my last previous paper on the same subject[2], I received so many comments both from individuals and from foreign and domestic governmental institutions that I felt the need of broadening my investigation. Many additional tests have therefore been made and, while the problem of airplane vibration is by no means solved, I may say that it is under control. A certain order also appears to have been established out of the chaos of ideas, the existence of which was shown by the attempt to bring a solution of the problem by the questionnaire method[3].

For the reader to follow the present paper, it is suggested that he acquaint himself with the previous one[2], as many math-

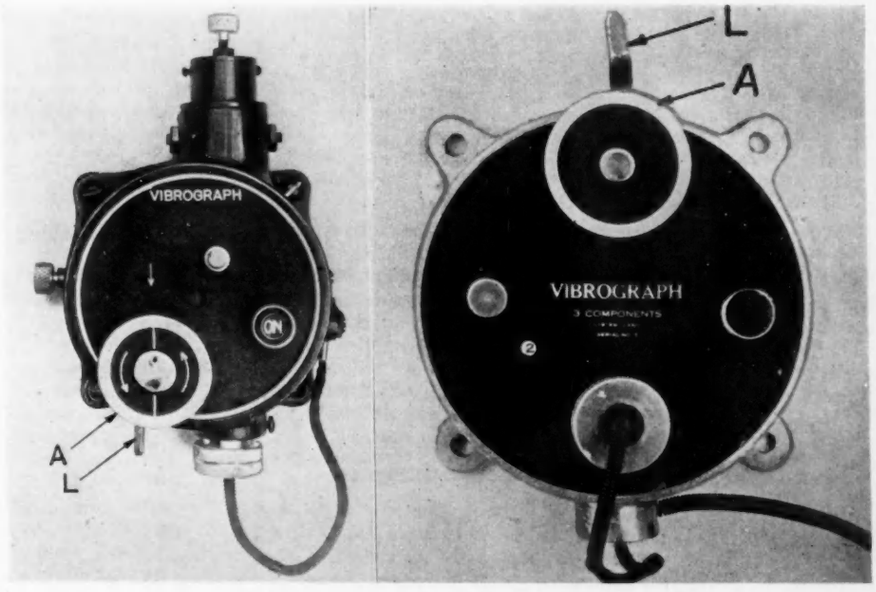

FIG. 1—VIBROGRAPHS DEVELOPED BY THE AUTHOR

Two-Component Vibrograph Built in 1931. *A*, Operating Knob. *L*, Safety Lever, Which Must Be Depressed Before Instrument Can Be Operated. The Knob above the Center Is Used To Set the Exposure Indicator

Three-Component Vibrograph, in Latest Form. Note Simplicity of Design. The Mechanism Is Contained in a Standard Aircraft-Instrument Case, the Dial Being 2¾ In. in Diameter, the Same as for the Two-Component Instrument

[1] M.S.A.E.—Consulting engineer, Forest Hills, N. Y., temporarily retained for special work in engineering division, Sperry Gyroscope Co., Inc., Brooklyn, N. Y.

[2] See S.A.E. JOURNAL, October, 1931, p. 263.

[3] See Report of the American Society of Mechanical Engineers on Airplane-Instrument Vibration, December, 1930.

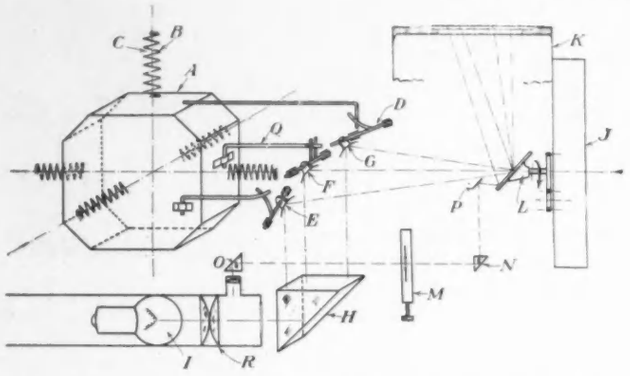

FIG. 2—SCHEMATIC DIAGRAM OF THE THREE-COMPONENT VIBROGRAPH

Parts Referred to Are: *A*, Tungsten Mass; *B*, Springs; *C*, Filler for Damping Springs; *D*, Hair-Springs; *E*, *F* and *G*, Spherical Mirrors; *H*, Reflecting Prism; *I*, Lamp; *J*, Clockwork; *K*, Film Drum, the Axis of Which Coincides with Axis of Revolving Mirror *L*; *M*, a Tuning-Fork or a Chronometric Shutter; *N* and *O*, Prisms and *P*, Mirror Reflecting the Timing Beam (Optional); *Q*, Transfer Mechanism and *R*, Lenses

ematical formulas and other data will be taken directly from it, thus avoiding duplication.

The first vibrograph built in 1930 served well for many preliminary tests but had a number of shortcomings that had to be corrected before an attempt was made to use the apparatus commercially. The difficulties were that:

(1) Operation of the vibrograph required great skill, and the motions necessary to make a record necessitated the simultaneous use of both hands. Thus, to obtain records on a single-seater airplane was out of the question.
(2) Loading of the vibrograph with film had to be done in a completely darkened room and required special film as well as a trained operator.
(3) Mounting of the instrument on an instrument-board was accompanied by several difficulties, and in most cases special adapters had to be used.
(4) Records obtained were much distorted, as the clockwork did not run with constant speed.
(5) Interpretation of the records was also difficult because the instrument recorded a resultant of two motions.

For the foregoing reasons the two-component apparatus was completely redesigned and several of the new instruments were placed in the hands of investigating agencies in this Country and abroad. The finished instrument, shown at the left in Fig. 1, is a compact piece of apparatus and all the deficiencies mentioned above, except (5), have been eliminated. In addition, completely automatic operation is incorporated; the operator has only to turn the knob one-half revolution and a record is obtained.

Design of a Three-Component Instrument

In the course of the investigation I realized that a two-component instrument has objectionable limitations. As some parts of the airplane structure vibrate in three directions, the use of two vibrographs was necessary to obtain a complete record. Further, these records would give a resultant vibration curve, either an *X*-and-*Y* or a *Y*-and-*Z* *combination* and considerable skill and tedious calculation were required to separate the components; on several occasions I failed to obtain an intelligent interpretation of the results because I could not separate the corresponding components.

After many failures I finally designed and built a three-component instrument which is shown at the right in Fig. 1, and this design has made all my previous vibrograph designs obsolete. This instrument consists of two major parts: (*a*) the seismograph and (*b*) the automatic recording device. I bore in mind when laying out the instrument that this apparatus should be versatile, and therefore arranged the design in such a way that it can be used not only on airplanes but wherever vibration of medium and high frequency occurs. Like the improved two-component apparatus, this recorder is fully automatic in operation.

The theory of pendulous and spring vibrographs as set forth in the previously published paper[4] applies equally well to a three-component apparatus. To obtain a low natural frequency of the instrument, a heavy mass is necessary. In the three-component instrument, which, like the two-component instrument, is contained within a $2\frac{3}{4}$-in. housing in the standard aircraft size, tungsten is used as a mass, the specific gravity of this metal being 18.77 and the price not prohibitive. Thus a cube of 1.5-in. size has a weight of $1.5 \times 2.25 = 3.375 \times 0.68 = 2.28$ lb., or slightly more than 1 kg. With this mass and suitably damped springs, the natural frequency of the instrument is sufficiently low to enable the recording, without applying any corrections, of vibrations of a frequency of 800 cycles per min. and upward. By applying a correction coefficient, vibrations of frequencies as low as 350 per min. have been measured successfully.

The diagrammatic drawing, Fig. 2, shows the principle of the instrument. A cube of metallic tungsten, *A*, is suspended in the center of the case by 12 springs *B* in such a way that it is free to move in any direction. All the springs are damped by filling them with an oil-impregnated material, *C*. Three mirrors are suitably mounted in the rear of the tungsten block and each carries a transfer mechanism of such characteristics that each mirror can move in only one direction. Small hair-springs *D* assure perfect contact with the block *A*. At the point of contact the block is highly polished and, owing to the small mass of the mirrors and transfer mechanisms, the errors due to friction are almost nil. The mirrors *E*, *F* and *G* are not flat but spherical and of such optical characteristics that the image of the filament, coming through a prism, *H*, from the lamp *I*, is reflected on a film that acts as a screen. The airplane recorder consists of a light plate that carries two film housings and three posts to which the clockwork is attached.

Operation of the Driving Mechanism

The name clockwork is perhaps a misnomer; the driving element resembles more a miniature gramophone motor, but, owing to its accuracy and the fact that through its use the frequency of the record is obtained, the name probably is justified.

Let us assume that the main spring *A*, Fig. 3, is wound up; then, if the lever *B* is depressed for an instant, the mechanism is freed and the gear *C* will make exactly one revolution, because at the end of the cycle the projection of the lever *B* at *D* will again fall into its mating hole *E*. As the main drive gear has a number of teeth cut out, the gear *F*, which carries the mirror *G*, will revolve, stop and revolve again. As the mirror is very light and the gear *F* is mounted in jeweled bearings, the force necessary to accelerate it is very small; if the clockwork at that particular moment revolves with constant speed, the mirror also will revolve with constant speed. To assure such motion of the clockwork, a number of intermediary gears, *H*, *I* and *J* are mounted, the last of which carries a rather heavy disc, *K*, which at full speed revolves at a rate of about 6000 r.p.m. A centrifugal governor, *L*, which increases or decreases the friction of the lever *B* against the disc at the point

[4] See S.A.E. JOURNAL, October, 1931, p. 268.

M, further increases the constancy of the speed. No exact data on the accuracy of this movement have been compiled, but experiments seem to show that this clockwork, if always wound up to constant tension of the main spring A, will be accurate within 5 per cent. Of course, we have to assume that the temperature remains fairly constant.

The fact should not be overlooked that the time during which gear C makes one revolution is about 7 sec. and the record is made in the sixth second during a time interval of only 0.35 to 0.45 sec., and hence constant speed of the mirror G can confidently be expected.

Wear and tear resulting from the abrupt stop after one revolution would be detrimental to the bearings of the gear C, as the speed is rather high. Therefore I have incorporated a brake that comes into action after the record has been made, that is, in the last ¼ sec. of the 7-sec. cycle. The disc in which the hole E is located is slightly flattened on one side; hence the lever B, before reaching the position shown in Fig. 3, moves upward, actuating a friction brake, N. As the shoe of this brake rubs against a wheel that has a high rotational velocity, a very smooth and effective deceleration is obtained. If, for some reason, the gear F should slip one or more teeth while it is out of mesh with the gear C, the cam O, similar to a centralizing cam of a stop watch, will assure that the next cycle starts with the same original relative position of the two gears.

Automatic Operation and Safety Features

This apparatus also is fully automatic and operation of the vibrograph requires only turning knob A, Fig. 1, one-half a revolution counter-clockwise. This knob is keyed to a shaft, the opposite end of which fits into the film take-up spool. Every half turn of the shaft will, therefore, move the film in relation to the gate by 1⅞ in. The same shaft carries a gear that, through an idler, engages the gear A, Fig. 3, which forms part of the main-spring-housing gear. The ratio of the gears is made such that the spring is brought back to the original tension after each release of the clockwork, which is made automatically at the end of the half turn of the operating knob. The shaft carries a cam P, which, through a system of levers, pushes the lever B downward. It can at once be seen that the light-switch is "on" only when the clockwork is running and that an exposure indicator is also operated by the cam P.

To recapitulate, one-half turn of the operating knob:

(1) Moves the film 1⅞ in., thus assuring proper spacing of the records
(2) Winds up the main spring to a predetermined and always constant tension

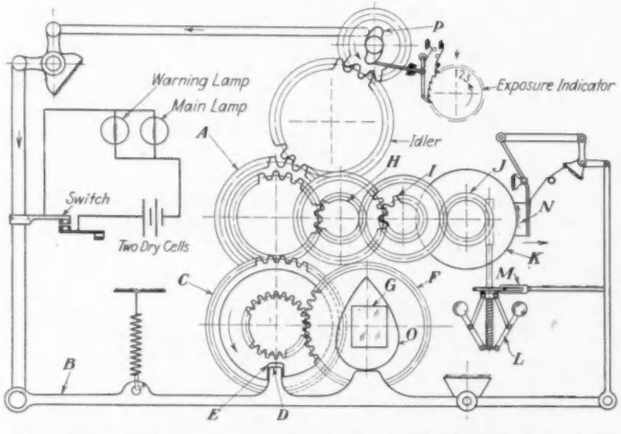

FIG. 3—SCHEMATIC DIAGRAM OF CLOCKWORK AND AUTOMATIC FEATURES OF THE TWO-COMPONENT AND THE THREE-COMPONENT VIBROGRAPHS

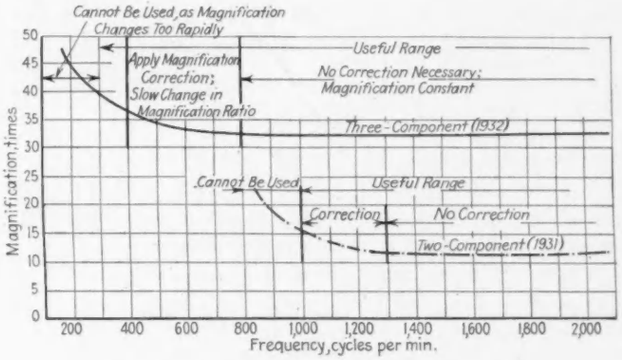

FIG. 4—VIBROGRAPH CALIBRATION CURVE

The Three-Component Instrument Gives Constant Magnification over a Range from 800 Cycles per Min. Upward, and by Applying a Correction Can Be Used All the Way Down to 400 Cycles per Min. The Magnification of the Full-Line Curve Is 33. The Dash-Dot Curve Is for the Obsolete Instrument, Which Has a Much Shorter Range, from 1300 Cycles per Min. Upward, the Magnification Being only 11

(3) Releases the clockwork at the correct instant
(4) Operates the exposure indicator
(5) Switches on and off the main light and the warning light
(6) Stops the clockwork smoothly after each exposure

As a certain relationship between the cam and the position of the mirror is to be maintained, the knob should be turned only 180 deg. Therefore a safety device is incorporated, and a lever, L, Fig. 1, must be depressed before the knob can be turned. This prevents the operator from making more than one-half turn at a time, assuring absolutely foolproof and automatic operation of the vibrograph.

To wind up the film leader or to take up the remainder of the film into the film magazines, the knob can be pulled out about ¼ in., automatically disconnecting the clockwork, the switch mechanism and the release mechanism. The knob can be clutched in only in the right position. This safety device makes impossible the reengagement of the clutch in any other position.

A centralizing and locking device also has been incorporated to assure safe transportation and avoid undue strain on the springs and transfer mechanism when the apparatus is not in use.

Daylight-loading magazines are used, and films can be changed in subdued daylight. About 20 records can be made with one loading. I have also made an instrument that uses standard vest-pocket verichrome film, with which only 12 records can be obtained, but the operator has the advantage that this particular brand of film can be obtained in any part of the civilized world.

Should the investigator be very particular about the frequency and not want to depend upon the constancy of the speed of the clockwork, a tuning-fork or a chronometric shutter can be incorporated. Its function can be seen clearly at M in Fig. 2. For different purposes a different recorder can be attached to the seismic part.

Fig. 4 shows the respective calibration curves of the two instruments, from which the superiority of the three-component instrument design over the two-component vibrograph can be seen. It has a wider range and its magnification constant is three times larger than that of the two-component instrument.

Vibrograph and Accelerometer Difference Explained

Vibrographs and accelerometers apparently are very similar in design and some confusion when referring to these two instruments seems to exist. I think the best

method of showing how they differ is to make a complete mathematical analysis. I refer here to Equation (21) in my previous paper[5] and to the conclusion we drew from this mathematical interpretation. We saw that

$$\alpha = (al/k_1^2) \cdot [q^2/(n^2 - q^2)]$$

This equation can also be written as

$$\alpha = (al/k_1^2)/[(n^2 - q^2)/q^2] \quad (1)$$

but al/k_1^2 is a constant of the apparatus, hence it can be called C and $(n^2-q^2)/q^2$ can be written as $(n^2/q^2) - 1$. Thus Equation (1) becomes finally

$$\alpha = C/[(n^2/q^2) - 1] \quad (2)$$

If the frequency n, which is the natural frequency of the apparatus, is small in comparison with the frequency of the impressed vibration q (say ratio 1:5 or better), then the denominator of the fraction will approach the value of -1 and we can say that $\alpha \sim C$; therefore the recorded amplitude is almost equal to the amplitude of the impressed motion, and, consequently, if we desire to record a motion, vibrograph, the apparatus must be built so as to have n much smaller than q. If, however, we design the apparatus in such a way as to have n much larger than q, Equation (2) can be written approximately as $\alpha = C(q^2/n^2)$, and, since in a given instrument n is a constant, the recorded amplitude is proportional to Cq^2; but the impressed motion is $C \sin qt$ and its acceleration $(d^2\alpha)/(dt^2)$ is $-Cq^2 \sin qt$, which is also proportional to Cq^2. Therefore, if we have n much larger than q, we record accelerations. Near the point of resonance or when n equals q, the instrument will be suitable neither for an accelerometer nor a vibrograph, as theoretically α would become an infinite value, disregarding damping.

From this study we may conclude that theoretically, in some cases, both vibrograph and accelerometer can be used for the same purpose. Unfortunately, the time element in which a single cycle of a vibration occurs is so much shorter than the time during which the airplane generally is accelerated that the speed of the film or mirror reflecting the light beam would have to be suitably adjusted in each case to obtain a legible record. I doubt if a two-purpose instrument for airplane use will ever be designed[6].

Analysis of Vibration Troubles on Selected Instruments

Resonance Condition.—The instrument engineer faces many problems that the airplane engineer does not know. Instruments have to be built in such a way as to withstand all kinds of conditions, and often the satisfactory solution for one requirement makes the instrument less suitable to meet the remainder of the specifications; hence compromises have to be made.

The main difficulty is always the small force available to produce an indication. In an altimeter, for example, for a difference in altitude of 1000 ft. the available power results from about 1 in. of mercury-pressure difference. For a range of 10,000 ft. the pressure difference will be only about 6 in., owing to the logarithmic form of the pressure-altitude curve. The movement of the sensitive element is about 1/32 in., or possibly 1/16 in. This small movement must be magnified by a system of levers or other arrangements about 100 or 200 times to be read with ease. If we consider a sensitive altimeter, one making one full revolution for each 1000 ft., we see that the problem is still more difficult to solve.

Let us now assume that we have built an altimeter or have established all component parts, such as diaphragm, spring and main arm, and are trying to make this particular design vibration-proof. Assume also that an airplane manufacturer complains that the instrument has too much error because of vibration of the hand. Fig. 5 shows at the upper left a simplified diagrammatic drawing of an altimeter. This is known among the instrument makers as the C-spring design and was once very popular with many manufacturers. We see that it really resembles a vibrating reed and consequently can be represented as in the middle diagram at the left. In calculating the natural frequency of this system we have to take into consideration the weight of the main spring and the main arm, as this mass will have an effect on the natural period. If one-quarter of this weight is imagined as being added at the end of the beam, the results, according to Prof. S. P. Timoshenko, will be correct within 1 per cent. The static deflection of the system will then be

$$\delta_{static} = [(W + 0.25\ W_1)\ L^3]/3EI \quad (3)$$

and the period of such a system is

$$p = 1/f = 2\pi\sqrt{(\delta_{static}/g)} \quad (4)$$

We have previously seen that, to prevent resonance phenomena, the natural frequency of the instrument should be not greater than one-fifth that of the impressed vibration. From previous tests and experiments we know that f equals n—if n is the operating speed—hence at an assumed engine speed of 1800 r. p. m. f equals 30 per sec. or, to satisfy the above conditions, $p = 1/f = 5/30 = 1/6$. If we substitute δ_{static} in Equation (4), its value, from Equation (3), will be

$$p = 1/f = 2\pi\sqrt{[(W + 0.25\ W_1)\ L^3]/(3EIg)} \leqq 1/6 \quad (5)$$

Assuming now that in Equation (5) the conditions are *not* fulfilled, we see that, to satisfy this non-equality, we shall have to change either W or W_1, or L or E, or lastly, I. But to change W, which depends upon the initial tention of the main spring, would mean also to change the useful range of the instrument, as this is governed by the stiffness of the main spring. To change W_1 would mean no changes in the altimeter performance, since, though the main arm usually is made of brass, it can be made of duralumin, leaving all dimensions unaltered. This will decrease W_1 by about two-thirds, but the effect on Equation (5) is small, as it changes only in proportion to $(W_1/4)^{-2}$.

To change L would mean a rather satisfactory solution as regards vibration, as L is in the third power; but the length of the main arm determines the initial magnification, and any change in this dimension may result in a very difficult, if not impossible, calibration. To change E, the modulus of elasticity, is impossible; very little choice is afforded in the materials to be used; and the moment of inertia I cannot be changed, as this would automatically change W and W_1.

From this simplified analysis we see that, once a C-type altimeter has been successfully designed to fulfill the requirements of range, calibration, temperature error and the like, if it does not meet the vibration specification, then improvements in this respect become very difficult and may even be hopeless. The whole arrangement of the C-type altimeter is not logical from a vibratory point of view, as the flexible parts lie in a plane perpendicular to the fore-and-aft axis of the airplane. As we have previously found out, instrument-board vibration is mostly parallel to the longitudinal axis X, hence the full effect—not components—of those vibrations is transmitted directly to the mechanism of the instrument.

A much better design is the one devised by Kollsman about four years ago and represented at the lower left in Fig. 5. This instrument does not employ a spring. The diaphragm itself acts as such and is located at right

[5] See S.A.E. JOURNAL, October, 1931, p. 268.

[6] See National Advisory Committee for Aeronautics Reports Nos. 99 and 100.

angles to the board. The main arm does not form a part of the spring but floats on the pivots at points *A* and *B*. Very little or virtually no vibration trouble has been found with this type. It is a logical design as regards vibration and, as it has other advantages from purely a production viewpoint, it probably will replace the C-mechanism in a short time.

Instrument Deterioration Due to Vibration

An airplane-instrument manufacturer not infrequently receives complaints that some instruments deteriorate prematurely. My experience has been that in almost all cases the cause is in the wrong mounting of the instrument in the airplane. In particular, I want to refer here to a very common instrument, the pressure gage. Cases are on record of pressure gages that have operated successfully for years deteriorating in a very short time on a particular three-engine airplane where they were mounted outside on the nacelles. I measured the amplitude of vibration at this particular location and found it to be very high, about 1 mm. By a simple mathematical calculation I shall try to show that, with this particular amplitude, assuming an engine speed of 1750 r.p.m., the mechanism of this gage, represented at the top right in Fig. 5, is stressed beyond the normal stresses allowed for the particular material of which it is made. As in any problem of this type, we have to consider: (*a*) the stresses due to static conditions and (*b*) the stresses due to dynamic conditions.

All calculations in the following analysis are in metric units.

The force *P* acting on the center of gravity of a tooth can easily be calculated, as a cross-section of the Bourdon tube and all the other elements are known for this particular design. The force is 0.85 kg. for this design. The dimensions of the tooth are given on the sketch at the lower right in Fig. 5. Then the bending stress in the tooth is $S_B = LP_s/Z$, where *L* is the height and *Z* is the section modulus, or

$$S_B = 0.0375 \times 0.85 \times 6/0.05^2 = 1560 \text{ kg. per sq. cm.}$$

This is absolutely normal, as bronze will withstand a bending stress of 5000 to 6000 kg. per sq. cm.; in other words, under static conditions the factor of safety is about 3 to 4.

Now let us submit this gage to a vibratory motion of 1-mm. amplitude at a frequency of 1750 per min., or 28 per sec., parallel to the *X*-axis of the plane, as in the case of the airplane under study. If a body of the weight *W* is subjected to a simple harmonic motion of the amplitude $2a$ and has a frequency of *f* per second, we must find out what forces give this mass this motion. This requires going through the following procedure:

If *X* is the displacement of the body from mid-position at any instant, we may take the motion to be $X = a \sin qt$ or $a \sin 2\pi ft$. Then the numerical value of the acceleration at any instant is $4\pi^2 f^2 X$. The force drawing the body to this mid-position is (in kilograms)

$$P_{dynamic} = 4\pi^2 f^2 WX/9.81$$

or in our case,

$$P_{dynamic} = 4\pi^2 \times 28^2 \times W \times 0.5/98{,}100.$$

The term *W* can be calculated as a reduced weight of the whole system and concentrated at the center of gravity of the tooth. Calculation of this value is simple in principle, but the numerical evaluation would take so much space that this part of the work is omitted. The value is, of course, very small and in our case is 3.5×10^{-6} grams.

Thus the dynamic force is

$$P_{dynamic} = 4\pi^2 \times 28^2 \times 1/2 \times 98{,}100 \times 3.5 \times 10^{-6} = 2800 \text{ grams}$$

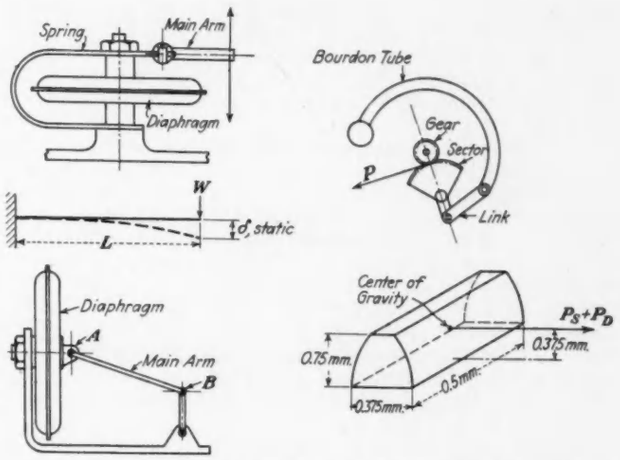

FIG. 5—DIAGRAMS OF ALTIMETER AND PRESSURE-GAGE PARTS

(Upper Left)—So-Called C-Spring Type of Altimeter

(Middle Left)—A Vibrating Reed with Which the C-Type Altimeter Can Be Compared

(Lower Left)—Kollsman Type of Altimeter, in Which the Spring Is Omitted, the Diaphragm Acting as the Spring. The Main Arm Floats at Points *A* and *B*

(Upper Right)—Diagram of a Pressure-Gage. If This Gage Is Subjected to an Abnormal Vibration, the Sector Becomes Worn in a Very Short Time

(Lower Right)—Dimensions of a Sector Tooth of the Pressure Gage, in Millimeters. Calculation Shows That This Tooth Will Wear Excessively If Subjected to a Vibration of Too Large Amplitude

As, however, $P = P_S + P_D$, in our case we will have

$$P = 2.800 + 0.850 = 3.650 \text{ grams}$$

Therefore the stress

$$S_B = LP/Z = 0.0375 \times 3.65 \times 6/0.05^2 = 6500 \text{ kg. per sq. cm.}$$

As best bronze can stand a stress of only 5000 to 6000 kg. per sq. cm., the gear sector is overstressed by 10 to 30 per cent during a vibration of 1-mm. amplitude at a frequency of 28 per sec; hence the sector will deteriorate after a short time, sometimes in two or three weeks. This difficulty arises from the mounting of the gage in the wrong place and also from too great a vibration of the nacelle. The designer of this particular airplane should either improve the outboard nacelles or design a special mounting for the gage, and this procedure will be discussed later. This type of gage is entirely satisfactory under normal conditions, and laboratory tests subjecting this gage to more than 500 hr. of continued vibration did not disclose any ill effect resulting from stresses developed by the so-called standard vibration.

Compasses.—The greatest number of difficulties are encountered with compasses. This is because the airplane compass really is a very difficult instrument to design. The necessity for having a liquid as a damping medium, the very small directive force available and the comparatively great weight of the instrument contribute to the poor performance of the compass under vibration. In the last four years, however, systematic research has been carried out, notably by the Army and the Navy, which resulted in the design of a compass that will withstand vibration equal to or even greater than that encountered on the usual airplane. Almost invariably, however, the compass is mounted through a shock-absorbing medium of some sort. Any compass can be made to perform satisfactorily provided the mounting is studied out very carefully. The most common trouble is

spinning; that is, the compass card starts rotating and continues to turn until the aircraft is brought to a stop. Another rather common occurrence is that the compass card will assume an angularity against the horizontal line, and indications against the lubber line are not true[7].

As an exact mathematical analysis of compass troubles apparently cannot be worked out, many experiments were carreid out with compasses, and some of the experiments yielded very interesting designs. For example, on the B-8 Army compass, a shock-absorber consisting of a spiral spring to which the pivot of the card is attached has been built into the card. This spring acts as an absorber in two directions. Results with this type are encouraging, but it is somewhat less satisfactory than is the Navy solution, which uses a pivot suspension on a light spring and a special suspension of the whole compass. This mounting will be treated elsewhere.

I have made innumerable experiments with instruments and also made a survey of complaints. These are tabulated in Table 1, which shows that the large majority of instruments will withstand vibration of much higher amplitudes than those encountered in the majority of airplanes. Therefore I must repeat the statement[8] that

> The failure of instruments in some airplanes can be directly attributed to the type of installation.

However, a great number of instruments, especially of foreign origin, are so far below the American standard that they have not been taken into consideration in this survey.

Decreasing Vibration of Existing Installations

As a universal cure for vibration, many engineers are advocating interposing springs or some other materials, such as sponge rubber, felt or other fibrous material, between the instrument-board and the part supporting it. Introducing the flexibility of springs or of flexible material, which can be regarded as the sum of an infinite number of small springs, between a machine and its foundation can achieve a substantial reduction in the force transmitted to the support *provided certain conditions are fulfilled.* If these conditions are not taken into consideration, the result may be that, not only is the vibration not *decreased*, but it is materially *increased.* This explains why I have always been so reluctant, in different discussions and papers, blindly to recommend springs or sponge rubber as a cure-all for vibration trouble. Unless one actually is familiar with the problems of forced vibrations with damping, grave errors may be made in trying to eliminate vibration by using springs or rubber supports. This is true, not only in the case of instrument-boards, but anywhere that the engineer has to cope with the absorbing of vibration. To illustrate, after a well-known manufacturer had introduced a new model of automobile in 1928, the car was known as very "rough." To eliminate the discomfort, the engineers of this particular car introduced in the engine support a spring absorber that presumably was designed by rule-of-thumb, with the result that more than 50,000 cars had to have their universal-joints replaced, as they had failed as a result of excessive vibratory stresses.

Yet the calculation of the effect of an absorber can be made very easily. While textbooks are full of formulas, I find that the material is often scattered. I shall try to give in logical sequence the development of these formulas. Proceeding very methodically is necessary to see the simplicity and usefulness of the mathematical calculation.

[7] See Fig. 1, S.A.E. JOURNAL, October, 1931, p. 263.
[8] See National Advisory Committee for Aeronautics Report No. 371, p. 21.
[9] See Calculus for Engineers, by J. Perry, 14th edition, Edward Arnold Co., London, England; see also, Vibration Problems in Engineering, by S. Timoshenko, D. Van Nostrand Co., New York City.

TABLE 1—MAXIMUM SAFE AMPLITUDE FOR DIFFERENT INSTRUMENTS AND TROUBLE EXPERIENCED WITH THEM

Instrument	Observed Trouble	Ax Maximum Safe Amplitude, Mm.	Complaints or Remarks
Altimeter	Hand vibration	0.55	Only one type affected; complaint from only one manufacturer
Sensitive Altimeter	Hand vibration	0.40	Not conclusive; only one type tested
Airspeed Indicator	Hand vibration	0.85	No complaints
Tachometer (Centrifugal)	Hand vibration and deterioration	0.70	Few complaints
Tachometer (Chronometric)	Deterioration and erroneous indications	0.45	Not conclusive; only two tests
Tachometer (Electric)	Hand vibration and deterioration	0.30	Not conclusive; few tests
Thermometer	Hand vibration and deterioration	0.65	Complaints from one manufacturer
Oil-Pressure Gage	Hand vibration and deterioration	0.70	Few or no complaints
Fuel-Pressure Gage	Hand vibration and deterioration	0.50	One case of bad installation
Compass (Magnetic)	Spinning, card angularity and deterioration	0.20-0.35 0.45	Depending on type; frequent complaints
Compass (Earth-Inductor Indicator)	Rapid deterioration	0.25	Almost all indicators break down in a short time
Compass (Telepoint)	Deterioration	No data	No data
Rate-of-Climb Indicator	Hand vibration	0.90	No complaints; best instrument
Clock (Spring)	Change in rate and deterioration	0.85	No complaints
Clock (Electric)	Change in rate and deterioration	1.00	No complaints
Artificial Horizon	Erroneous indications	0.60	No complaints
Directional Gyro	Erroneous indications	0.80	No complaints
Turn Indicator	Hand vibration	0.90	No complaints
Bank Indicator	Erroneous indications	0.70	Complaints frequent

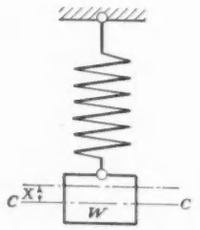

FIG. 6—DIAGRAM ILLUSTRATING THE MATHEMATICAL ANALYSIS OF SPRING VIBRATION

Consider a spring (Fig. 6), at the extremity of which a weight, W, is suspended. The weight W includes the portion of the weight of the spring which contributes to the deflection. The stiffness of the spring is such that a force of one unit elongates it h units. When the weight is at CC, it is X units below its position of equilibrium. The forces urging it into the position of equilibrium are X/h, and as the moving mass is W/g, the equation of motion is

$$\frac{W}{g} \cdot \frac{d^2X}{dt^2} + \frac{X}{h} = 0 \qquad (6)$$

This is the fundamental differential equation of a natural vibration, and its solution can be found in any handbook[9].

Now imagine the same layout, but in addition that the body is being retarded—damped—by a force proportional to its velocity, or by $b(dX/dt)$. This force acts in the same direction as X/h; that is, always toward the position of equilibrium. Hence we may write:

$$\frac{W}{g} \cdot \frac{d^2X}{dt^2} + b \cdot \frac{dX}{dt} + \frac{X}{h} = 0 \qquad (7)$$

This is the equation of damped vibration. Assume now that the scheme is not changed, but in addition the point of support performs a periodic motion in itself and follows a law that can be written as $Y = f(t)$. Thus

$$\frac{W}{g} \cdot \frac{d^2X}{dt^2} + b \cdot \frac{dX}{dt} + \frac{X}{h} = f(t) \qquad (8)$$

The equation represents forced vibration with damping and can be integrated no matter how complicated the function $f(t)$ is, provided, however, that it is a periodic function. In fact, if $f(t)$ is a periodic function, it can

at once be developed into a simple Fourier series of sines or cosines. For our investigation we shall consider only the first term. Thus we have a perfect right to write $Y = f(t) = A \sin kt$, and then Equation (8) can be written as

$$\frac{W}{g} \cdot \frac{d^2X}{dt^2} + b \cdot \frac{dX}{dt} + \frac{X}{h} = A \sin kt \qquad (9)$$

As is usual in the solution of differential equations, a transformation has to be accomplished before a solution can be found. Multiplying the equation by g/W and calling $bg/W = 2D$, $g/hW = B^2$ *and* $Ag/W = a$, Equation (9) is brought to a "classical" form

$$\frac{d^2X}{dt^2} + 2D \cdot \frac{dX}{dt} + B^2X = a \sin kt \qquad (10)$$

Without any further consideration, we see that we already studied this equation when we attempted to analyze the problem of resonance between engine and engine mount. In the former case we took under scrutiny only one solution. Another solution will be taken into consideration now. The amplitude of the forced vibration is

$$\delta_{stat} \Big/ \sqrt{\left[\left(1 - \frac{K^2}{B^2}\right)^2 + \frac{4D^2K_1^2}{B^4}\right]}$$

or, using Timoshenko's method[10] of expressing such formulas, we have

$$C = \delta_{stat} \Big/ \sqrt{\left[\left(1 - \frac{\omega^2}{\omega_1^2}\right)^2 + \frac{\omega^2\gamma^2}{\omega_1^2}\right]} \qquad (11)$$

Where δ_{stat} is the static deflection and $a/B = (Ag/W) \cdot (Wh/g) = Ah$; $\omega = 2\pi/B$ is the period of the natural vibration of the system; $\omega = 2\pi/K$ is the period of the disturbing periodical force $A \sin kt$; and $\gamma = 2K_1/B$ is the quantity depending upon the magnitude of damping forces.

Neglecting for an instant the term $\omega^2\gamma^2/\omega_1^2$ from Equation (11), we can conclude that a forced vibration of the body W is synchronous with the motion of the support, its amplitudes being $1/\sqrt{[1 - (\omega^2/\omega_1^2)]} = \alpha$ times that of the point of support. As we know the lowest impressed frequency, we can always find a suspension of such characteristics as to make α *a fraction*, thus decreasing the amplitude of the board to whatever figures we find practical. But should we proceed blindly, enormous magnification of the amplitude may occur. If, for example, $\omega/\omega_1 = 4$, then $\alpha = 1/15$; or an initial amplitude of, say, 1 mm. can be decreased by proper suspension to 1/15 mm., which will be hardly noticeable and not at all likely to be destructive. To otain these conclusions, we have neglected the term $\gamma^2\omega^2/\omega_1^2$; γ being a figure that is very difficult to evaluate. As a whole fraction is additive to the denominator, it can only make α smaller, so that the knowledge of it is not important, especially when applying this formula in contemplating the use of springs as shock-absorber media. When using rubber or other plastic materials, where the factor γ is not under such perfect control as in a spring, this value becomes important.

Applying the aforementioned theory to the suspension of instrument-boards, we see that we must know only the values of ω and ω_1 to work out a suitable support. The previous investigation revealed that in most cases ω_1 is the engine frequency. The natural frequency of the board is ω, and to calculate it may present some difficulty, as this is a statically undetermined problem. Finding it by an experimental method is much simpler. The following scheme can be used to find this value quickly and with sufficient accuracy.

Mount the instrument-board with its instruments on a vibration stand and keep the forced amplitude constant, say ½ mm. Mount a calibrated vibrograph, instead of one of the instruments, preferably at the center. Change the frequency in increments of 25 or 50 cycles per min. and plot the recorded amplitude as in Fig. 7. The peak of the curve is the natural frequency of the board. Should a vibrograph not be available, a small scratch on the board can be observed through a measuring telescope, and if a stroboscope is available this second method is somewhat faster.

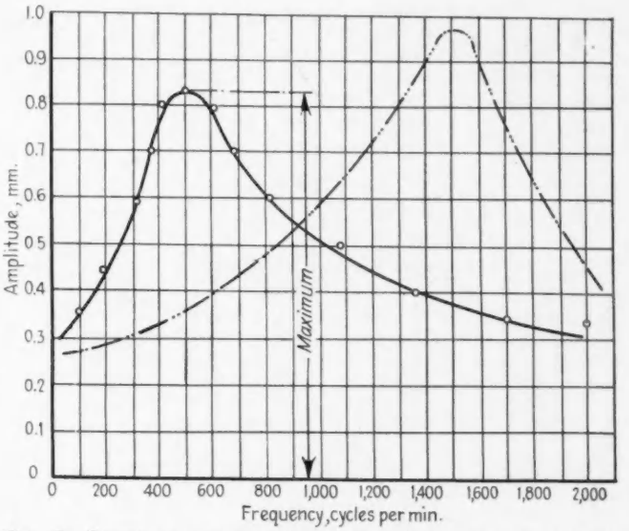

FIG. 7—DETERMINATION OF NATURAL FREQUENCY OF AN INSTRUMENT-BOARD

The Solid Curve Gives an Example of a Suitable Board. The Natural Frequency Is Very Low, Being 500 Cycles per Min., Which Is Outside the Flying Range. The Dash-Dot Curve Exemplifies a Board That Is Unsatisfactory Because Its Natural Frequency Lies in the Operating Range, That Is, about 1500 Cycles per Min.

The rest of the problem lies in the finding of a spring or other suitable medium giving an appropriate value to δ_{static}, which of course may present purely design difficulties. Low-frequency springs must be made of thin wire and have a large diameter; therefore, while the spring frequency is correct, the spring itself may not be safe, as it may be overloaded. No general rule can be laid down for a shock-absorbing design; each installation presents a special problem, but I want to emphasize that it is absolutely within the means of the engineer. In general, he should remember that:

(1) Instrument-boards should be made as stiff as possible, using angle or channel sections to stiffen the center part of large boards, to avoid flexural vibrations of the board itself. Under no circumstances should the board be made less than 1/16 in. thick, if using duralumin, or 3/32 in. if using half-hard aluminum.

(2) When supporting the instrument-board between springs, the natural frequency in line of the spring is not dependent on the initial tension of the spring. Thus varying the stiffness, and hence the natural frequency, along the spring axis by varying the spring tension is impossible. The spring constant has to be changed.

(3) When making experiments to establish the value ω, the natural period of the board, all instruments should be in their respective mountings and all connections, such as tubings and shafts, should be in place. The tachometer should be running at a speed corresponding to the frequency.

(4) Steel springs are not recommended, as they may badly affect the indications of the compass. Brass, bronze or monel metal is more suitable for this purpose.

[10] See Vibration Problems in Engineering, by S. Timoshenko, D. Van Nostrand Co., New York City.

Plastic Materials as Shock-Absorbers

If springs cannot be used, plastic or other materials can be substituted with great advantage, to decrease the forced amplitude. Great care should be taken in the

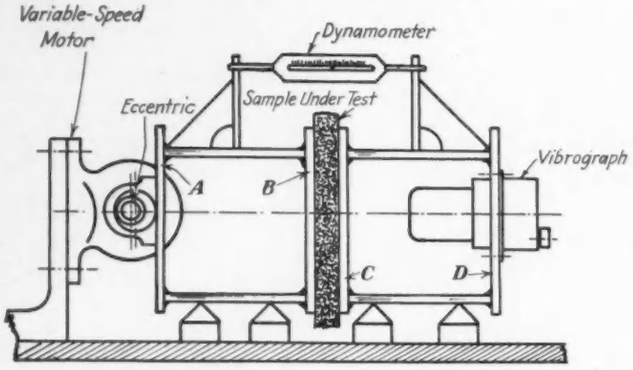

FIG. 8—DIAGRAM OF A TESTING OUTFIT USED IN ASCERTAINING SHOCK-ABSORBING QUALITIES OF SPONGE RUBBER AND OTHER PLASTIC MATERIALS

selection of such materials, for, if the material chosen is not suitable, increase of amplitude and introduction of harmonics will occur. When using plastic materials such as felt or cork, we depend largely upon the value of γ; and, to my knowledge, manufacturers of such materials have not as yet given this figure any mathematical or arbitrary expression.

The term "sponge rubber" means today just as much as "steel." Not having any nomenclature at hand, I shall refer to sponge rubber, as I obtained it from different manufacturers, as *A*, *B*, *C*, *D* and so on. Using all of these materials, we must take into consideration such variations as deflection per pound per unit of area, temperature and humidity during the time the material is used as a shock-absorber. I have not had opportunity to do any theoretical work in connection with plastic or synthetic materials, but a number of very excellent records have been obtained. Four are reproduced to show clearly what a vast difference in shock-absorbing qualities we may find in different materials.

The test apparatus represented by Fig. 8 consisted of two pairs of plates, *A-B* and *C-D*. The two plates in each pair were connected together with brass tubes and were hard-soldered. On one end a variable-speed motor gave the assembly, through a ball bearing and an eccentric, a simple harmonic motion of known characteristics. The distance of the assembly *C-D* from *A-B* was such that a piece of the material under test could be slipped between *B* and *C*. The tension thus obtained was measured by a simple spring dynamometer and kept constant for a given material. The vibrograph was placed at the end of the assembly *C-D*. The whole apparatus, except the motor, was kept enclosed in a wooden box in which the temperature and humidity could be measured and kept under certain control. To get the initial conditions, a piece of brass was inserted and the assembly clamped together; thus the whole assembly acted just as a rigid body would and the vibrograph record gave the initial amplitude and frequency with which the other materials were compared. The frequency on the records that are reproduced was 1800 cycles per min. and the original amplitude was 0.18 mm. The curve obtained is very nearly a picture of a simple harmonic motion, as shown by *A* in Fig. 9. Any departure from it is due to the secondary vibration of the building and the platform. As all these conditions remained unaltered during the tests, they need not be analyzed closely.

From the record of sponge rubber shown by Curve *B* we see that, instead of damping the original vibration, the amplitude was *increased* by about 40 per cent; that is, the system would be smoother if no shock-absorbers were used. Further, the double number of changes in the direction of the time-amplitude curve *B* at point *x*, *y*, *z* and so on, is due to acceleration, which will increase the deterioration of any instrument that may be attached to such a system. This is a typical case in which the wrong material has been used. Under different conditions this material may be excellent. The particular form of this curve can be presumed to be due to the fact that this sponge rubber has a rate of return to its normal thickness which is much too fast to act as an absorber, and consequently the impressed vibration combines with the first cycle of damped wave, giving as a result an increased amplitude. A damping of the motion follows, but the time for this process is too short, as the next forced impulse is already coming. This explanation is merely a justified guess, as obviously only an analysis of the harmonics would give us the complete picture of those conditions.

Felt was tested under similar conditions, and this material decreased the amplitude by 10 per cent, as shown by Curve *C* in Fig. 9; yet the same "heterodynic" effect in somewhat less extent can be seen. Comments similar to those on the previous test apply here.

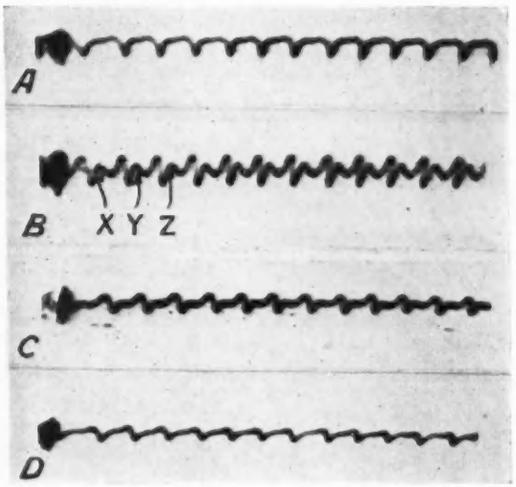

FIG. 9—EFFECTS OF SHOCK-ABSORBING MEDIA ON A SELECTED VIBRATION

No Shock-Absorber (Initial Condition) Shown at *A*; Sponge Rubber, at *B*; Felt, at *C* and a Synthetic Product at *D*

Still another material tested was one that I canno describe, as it is being sold commercially under a trade name. The result of the test is shown by record *D* in Fig. 9. Here the amplitude was decreased to about 30 per cent and the form is almost a pure sine curve. This was the right material to use under the conditions.

In the course of the experiments many different rubber and gelatinous synthetic materials were tested, the latter group proving to be more suitable for high-frequency vibrations than the former.

The effect of temperature on the selected materials has been studied and the results are shown in Fig. 10. As can be seen, sponge rubber, *B*, is suitable only at medium-high and low temperatures. Again, felt, *C*, is less effective at high humidity and more effective at lower relative humidities. Various gelatinous synthetic products have different characteristics also, but change in temperature or humidity does not seem to affect their shock-absorbing qualities.

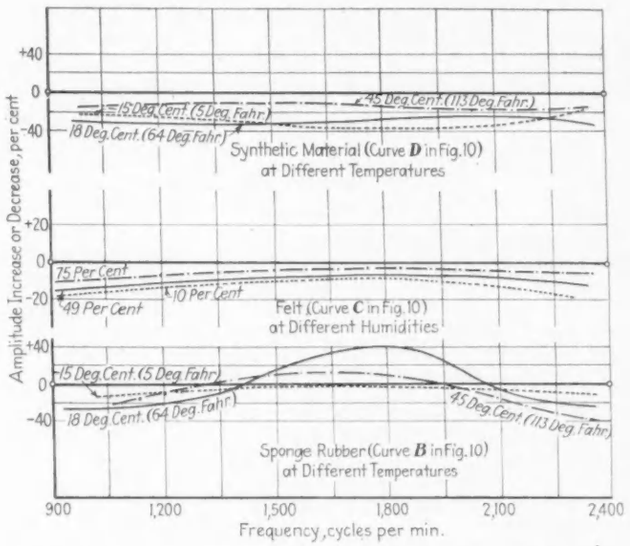

FIG. 10—EFFECT OF TEMPERATURE AND HUMIDITY ON SHOCK-ABSORBING QUALITIES OF SELECTED MATERIALS

All of these tests show rather conclusively that, unless great care is taken in the selection of materials, only doubtful results can be obtained. This is true not only of airplanes but also, as previously mentioned, of other moving vehicles. Again a test seems to prove my statement. I made a vibrograph test on a car advertised as having the engine suspended in rubber at eight points, measuring the amplitude at the engine support and then at the frame. The amplitude of the engine proved to be 0.29 mm.; that of the frame, 0.28 mm. The form was so identical and the difference in amplitude so small that it can be inferred that a solid connection would have been just as good. Apparently the vibrograph could be used to advantage in the automobile industry as well as in the rubber industry, giving the interested manufacturers indisputable records of vibration-absorbing qualities of different materials under different conditions.

[11] See Theory of Sound, by Lord Rayleigh, London, 1894.

[12] See Aircraft Vibration, by H. Constant, *The Journal of the Royal Aeronautical Society*, March, 1932, pp. 205-207.

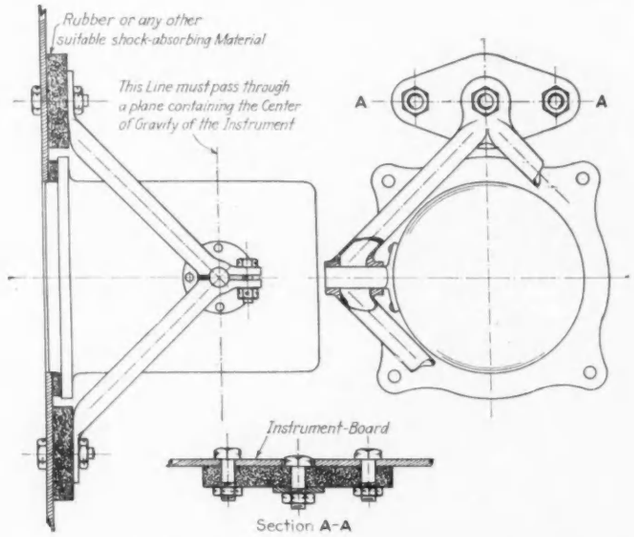
Rubber or any other suitable shock-absorbing Material
This Line must pass through a plane containing the Center of Gravity of the Instrument
A
A
Instrument-Board
Section A-A

FIG. 11—DIAGRAM OF A FLOATING INSTRUMENT-MOUNTING

Should an instrument-board vibrate in such a way that only one or two instruments are affected, insulating the particular instrument rather than the whole board would seem reasonable. Compasses and some gyroscopic instruments are not nearly so immune to vibrations as are other aircraft instruments. The mounting shown in Fig. 11 will prove very effective. Here the weight of the instrument is taken by a pair of braces. Further, the shock-absorber has a chance to flex and, last, all forces and moments must pass through the center of gravity of the system. The last item is the most important, as

> *any part of a motion which necessitates the motion of the center of gravity of the support is very quickly damped out of existence*[11].

We have given this system another degree of freedom and can show that in some cases this is the best solution of vibratory problems. Similar thought was applied in designing the Mark VI and Mark VII compass of the United States Navy, in which the compass bowl has a certain degree of freedom and the vibrations are damped out by highly polished friction washers. In this compass the center of gravity has an independent freedom and the result is that this instrument performs excellently under the worst vibratory conditions. "Floating power" is another example of the application of this method to get rid of the effects of excessive vibration.

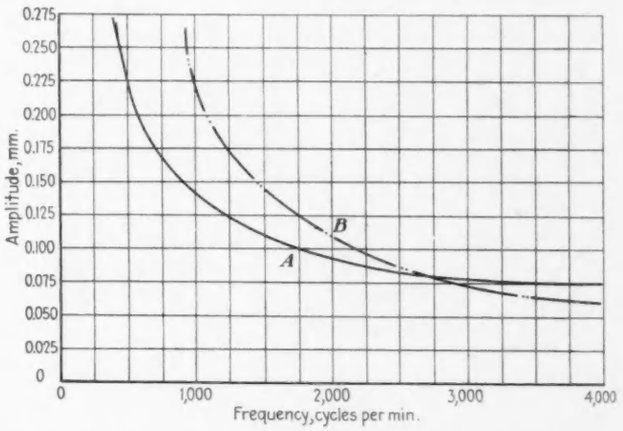

FIG. 12—RELATION BETWEEN AMPLITUDE AND FREQUENCY MARKING THE LIMIT AT WHICH A PERSON WILL NOT FEEL UNPLEASANTNESS

Curve *A* Was Obtained by H. Constant in England and Is Reproduced by Courtesy of the Royal Aeronautical Society. Curve *B* Was Obtained by the Author with the Cooperation of Dr. F. A. Moss, of the George Washington University

Vibration and Fatigue

As previously explained, before this investigation vibration was measured largely by bodily sensation. Most investigators have resorted to this method; for example, a series of tests were made in England where the test apparatus[12] was

> a simple contrivance whereby an observer could be subjected to a vibration whose amplitude and frequency could be altered at will. This consisted of a wooden beam hinged to a heavy bedplate at one end and mounted at the other end on an eccentric driven by an electric motor. The amplitudes of the beam thus varied from 0 mm. at one end to a maximum of about 2.5 mm., or 0.10 in., at the other end. An observer was seated on the beam and subjected to vibrations of various amplitudes and increasing frequency, and attempted in each instance to state when the amount of unpleasantness reached an arbitrary level

fixed by himself. It is obviously extremely difficult to obtain consistent results from such an unreliable measurement of unpleasantness. The results of making a large number of observations, however, have been taken as giving a rough indication of the variation of the effect involved.

Curve *A* in Fig. 12 gives the results of these English tests[13]. From the data I have collected, I compiled Table 2 of my previous paper[14], and, after reading the British report, I reworked my own results obtained during actual flight and plotted them over those obtained by H. Constant. That the curves, while not identical, are of the same nature is not surprising. The fact that the human body will withstand without undue fatigue either a high-amplitude low-frequency vibration or a low-amplitude high-frequency vibration has long been a tenet of psychologists. The curves thus obtained have a parabolic form which probably is due to the fact that forces created by vibration are proportional to the square of the frequency.

In the last year and a half great advance in the technique of measuring fatigue has been made through the unceasing efforts of both Dr. Moss and Hunt[15], who developed the wabblemeter and the visiometer. These instruments[16] make possible the measurement of fatigue in terms of arbitrary yet well defined indexes, thus entirely eliminating the human opinion which is the most difficult factor to keep under control. Owing to lack of funds for extensive tests, I can report only one test, which was conducted with the cooperation of a well-known airline. Vibrations of the passenger seats of the particular airplane studied were measured with a vibrograph and plotted against engine speed, as in Fig. 13. I found that the seats vibrated about 35 per cent more at 1650 cycles or engine revolutions per minute than at 1850 cycles.

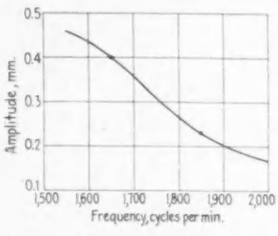

FIG. 13—RELATION BETWEEN FREQUENCY AND AMPLITUDE OF VIBRATION OF THE SEAT OF AN AIRPLANE

Four passengers were tested and their respective wabblemeter readings noted at the beginning and the end of the trip, which lasted exactly 1 hr. and was made at approximately 1650 r.p.m. The return trip, after a ¾-hr. rest, was made at 1850 r.p.m. The passengers' indexes were again recorded before and after the trip. The data are tabulated in Table 2 herewith, and the figures clearly show that fatigue increases almost proportionally with the amplitude. Correction for "frequency contribution" in this case was not necessary, as the curve in Fig. 13 follows somewhat that in Fig. 12. Other factors such as noise, odors and bumpiness also contribute to fatigue, but in our test an attempt was made to keep these other factors constant. If a much larger number of persons were tested, the results would be much nearer the truth, and I hope that such tests will be continued, as they certainly are very encouraging. One fact stands out very prominently; namely, that the wabblemeter and the visiometer are valuable instruments for both the engineer and the psychologist, giving indisputable figures and not opinions with which to deal.

[13] Curve *A* is reproduced through courtesy of the Royal Aeronautical Society.

[14] See S.A.E. JOURNAL, October, 1931, p. 278.

[15] Respectively professor and assistant professor of psychology, George Washington University, City of Washington.

[16] See S.A.E. JOURNAL, January, 1930, p. 99; April, 1930, p. 513; May, 1931, pp. 579 and 580; September, 1931, p. 243; and April, 1932, p. 182.

[17] See *The Journal of the Royal Aeronautical Society*, March, 1932, p. 218.

[18] See National Advisory Committee for Aeronautics Reports Nos. 99 and 100.

TABLE 2—READINGS FROM FATIGUE TESTS OF AIRPLANE PASSENGERS AT TWO AMPLITUDES OF VIBRATION

Passenger	Wabblemeter: Normal Score	Wabblemeter: Score after Flight	Wabblemeter: Increase	Visiometer Score
	Rate, 1650 *R.P.M.*; *Amplitude*, 0.4 *Mm.*			
A	27	38	11	5
B	33	40	7	0
C	19	29	10	7
D	24	36	12	10
	Rate, 1850 *R.P.M.*; *Amplitude*, 0.29 *Mm.*			
A	29	35	6	3
B	31	39	8	1
C	22	33	11	5
D	25	35	10	6

Sources of Airplane Vibration

The three principal sources of airplane vibration are: (*a*) aerodynamic disturbances, (*b*) vibration transmitted from the engine and (*c*) vibration caused by the propeller. Each is really a problem in itself; records obtained during flight give us the combined effect, hence any analysis of each separate item is very difficult. Yet this has been admirably worked out by Mr. Constant[17], and most of what follows is an interpretation of his work.

Any vibration results from a change in forces; aerodynamic forces change very often in a given airplane, hence tail flutter and wing flutter will be caused by the effect of the slipstream, change in course, change in pitch and the like. Only one thing is probable about vibrations of the aforementioned character; it is that they will be of low frequency and short duration. To get at least a rough idea of the magnitude of this type of vibration, several tests were carried out. Unfortunately, the records obtained were very irregular and almost impossible to analyze. One test, however, was successful, and the record is reproduced as (1) in Fig. 14. This vibrogram was made while the aircraft was sideslipping with the engine idling. Its amplitude was about 30 per cent greater than the average amplitude obtained in level flight, and the frequency was about 700 cycles per min. The vibrograph was mounted on the instrument-board and all records obtained in level flight were parallel to the *X*-axis, but in the case of the aircraft slipping, the plane of vibration seemed to be moved at least 10 deg. from the true *X*-axis. This is natural, as during the maneuver the plane will assume an angularity toward the perpendicular. Some tests were made with a dead stick to ascertain the natural predominant vibration of the plane. Several records were made with the vibrograph mounted on different parts of the structure. The frequency was computed for each plane and taken as the average of at least six readings; the results are given in Table 3.

Apparently, therefore, the frequency of a gliding airplane is inversely proportional to its total weight. These experimental results seem to be in agreement with the results[18] obtained in 1921. The authors of the reports on those results concluded, however, that the vibration is constant and always independent of engine speed, which seems to disagree with all recent experimental results. This discrepancy can be explained if one considers that the early experimenters lacked suitable instruments for making such delicate measurements.

TABLE 3—RELATION OF WEIGHT OF AN AIRPLANE AND FREQUENCY OF VIBRATION WHEN GLIDING WITH ENGINE STOPPED

Airplane	Weight, Lb.	Frequency, Cycles per Sec.
A	8,635	8.7
B	7,221	7.7
C	4,571	13.9
D	12,987	6.2
E	9,503	9.2

Classes of Engine-Induced Vibration

Vibration caused by the engine can be divided into two classes; (*a*) that arising from unbalance of rotating and reciprocating parts and (*b*) that resulting from changes in torque reaction. Let the engine designer study the forces and moments causing primary and secondary unbalance. A considerable amount of work has already been done, but the field is still open for a thorough analysis and many improvements. This statement applies especially to radial engines, in which seemingly balancing of the master-rod assembly gives considerable trouble.

H. Constant is responsible for the introduction of the idea of *specific unbalance*[19]. He says:

> For the purpose of comparing the relative inherent tendency of different engines to vibrate, the idea of specific unbalance has been developed. The specific-unbalance force of an engine has been taken to be the *unbalance force per unit of weight*, while the specific-unbalance couple has been taken to be the couple divided by the moment of inertia of the engine about an axis through the center of the mass perpendicular to the crankshaft axis.

This is decidedly a step in the right direction, as with these values vibratory limits of an engine can be impartially established. If the specific unbalance of an engine is known, the designer can at once correlate this value with the amplitude of vibration in the cockpit or cabin. Should this value exceed the limits shown in Fig. 12, he can, without actually building the airplane, suspend the engine in such a way as to reduce the amplitude to an acceptable figure. No work utilizing this expression has been done in this Country, and I sincerely hope that Mr. Constant's work will give a stimulus to designers to study this problem from this viewpoint.

The second source of unbalance—change in torque reaction—is a phenomenon that is so inherent in internal-combustion engines that the only way to get rid of it is to stop the engine[20]. For a given engine speed the values of the forces causing this unbalance are readily calculable and depend solely upon the speed. The moments due to torque reaction at a given speed depend, however, upon throttle opening, spark setting, firing order and so on, and this subject seems to be beyond the scope of this paper. However, as torque reaction is also a function of distribution, obviously a supercharged engine will, everything else being equal, be smoother than one that is not supercharged, as has long been recognized.

Origin of Propeller Disturbances

Vibration caused by the propeller alone may have its origin in (*a*) static unbalance, (*b*) blades out of track, (*c*) out of alignment or (*d*) out of pitch. These disturbances are rather easily recognized and remedied. However, the effects of cross-wind[21] and flutter and the gyroscopic effect in some cases can cause considerable vibration. The last two disturbances will be absent in level flight and should be considered only in an investigation of military airplanes, as civil aircraft are seldom subjected to violent maneuvers.

I was asked in the discussion of my previous paper if geared engines are smoother than direct-driven engines, and consequently have done a considerable amount of test work on both in-line and radial geared engines, using two-blade and three-blade propellers. Records from geared-engine installations are reproduced in vibrograms 2 to 5 in Fig. 14. These were taken on different makes of engine at different speeds and show one very

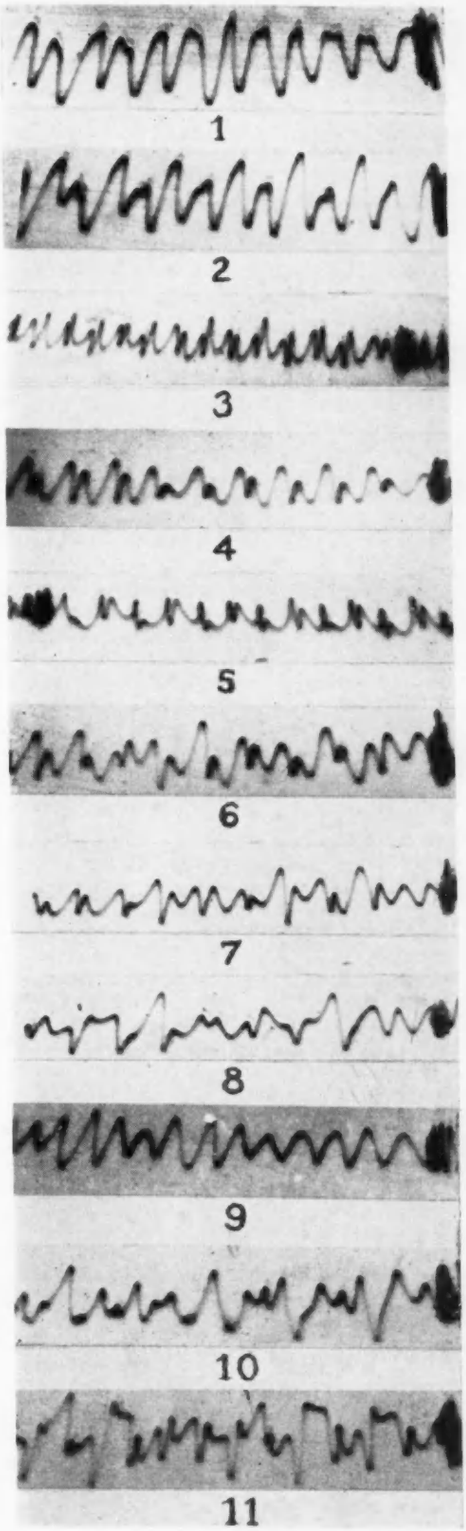

FIG. 14—VIBRATION RECORDS MADE UPON REPRESENTATIVE AIRCRAFT

[19] See *The Journal of the Royal Aeronautical Society*, March, 1932, p. 218.

[20] See Report of the American Society of Mechanical Engineers on Airplane-Instrument Vibration, December, 1930; see also Torque Reaction and Vibration, by H. Constant, published in *Aircraft Engineering* (British), June, 1932, p. 146.

[21] See Aircraft-Propeller Design, by F. E. Weick, 1930, McGraw-Hill Book Co., New York City.

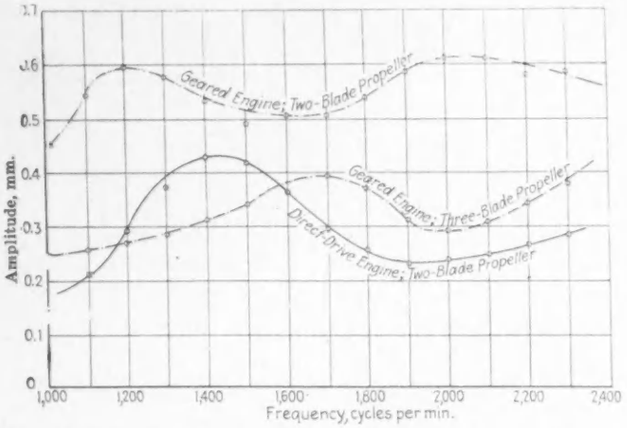

FIG. 15—CURVES REPRESENTING VIBRATION OF A DIRECT-DRIVE AND A GEARED-TYPE ENGINE WITH TWO-BLADE AND THREE-BLADE PROPELLERS

marked characteristic; that is, the very typical second harmonic[22]. Harmonics of higher order are present but are negligible. In most cases of similar installation the geared engine with a two-blade propeller gives amplitudes about 30 to 40 per cent greater than the direct-drive engine gives. The relation when using a similar airplane and exactly the same type of engine, either direct or geared, with different propellers is plotted in Fig. 15. The explanation of this phenomenon seems to be that in a geared engine we will have an increase in the harmonics due to torque reaction (Record 2, Fig. 14, is very typical) and also a lowering of the frequency of all forcing impulses due to the propeller.

The fitting of a three-blade or a four-blade propeller seems to decrease vibration considerably and the records obtained upon representative aircraft seem to prove this statement[23]. (See Records 6 and 7 for two-blade and 8 and 9 for three-blade, in Fig. 14.) Records 10 and 11, taken from a three-engine all-metal aircraft, show the effect of the three engines being slightly out of synchronism. They are very typical records, especially No. 11, showing "beating phenomena." These phenomena are, in my opinion, very dangerous and most unpleasant to the passengers.

The common notion is that vibration decreases with distance from the engine. I was under this impression myself, as in my previous work one of the amplitude-frequency curves[24] seemed to indicate that this belief is true. More accurate tests made at many points along the axis of the airplane show that this is not always the case. The curves in Fig. 16 show the relation. Naturally, a fuselage, being flexible, will have several nodes, and a vibration will be at the minimum at the nodal and at the maximum at the anti-nodal points. The three-component vibrograph is particularly suitable for a complete wing or fuselage analysis. Considerable test work would be necessary to formulate any definite conclusion regarding the flexural vibration of the fuselage and wing structure. It can be done, however, and probably will give most useful data.

Miscellaneous Uses of the Vibrograph

In the course of my experiments, a very interesting test was made in connection with aerial photography. If the mount of the camera vibrates excessively during the exposure, the pictures become blurred. To ascertain what speed of the shutter to use, the vibrograph was mounted on top of the camera and the amplitude and frequency were measured. These figures give the displacement per second of a point on the camera; from this the angular motion of the center of gravity can be calculated with ease. Having obtained this figure, a shutter speed can be chosen which will be much faster than this displacement. Should the calculation show that such a speed is not practical with existing equipment, the mounting of the camera must be changed. This information is given merely to illustrate that by use of the vibrograph many unnecessary experiments can be avoided and the solving of a vibratory problem becomes a matter of taking a few photographs.

Future Investigation

Mr. Constant's paper, my previous paper and this one do not exhaust the subject. Many facts remain to be found out. Some of the problems undoubtedly will be solved by individuals in subsequent researches. However, two very important items need the cooperative effort of the whole industry. I shall briefly outline what I have in mind.

Vibration is a nuisance to passengers and generally a dangerous phenomenon. While vibration can be given quantitative expression and in some isolated cases the forces due to vibratory disturbances can be correlated, nobody, to my knowledge, has approached the problem from the viewpoint of correlating the deterioration of materials with the amplitude and frequency of vibration and the *time* during which the material is subjected to such vibratory motion. Briefly, a structure made of a known material will, when new, withstand a certain load. This structure is then flown for a given length of time, during which it is subjected to a vibration of known amplitude and frequency. I wonder if, after such vibratory treatment for, say, a year or so, the material will withstand the same stresses as when new. I sincerely doubt it and think that crystallization of materials in airplane structures has been and is the cause of many accidents attributed to "unknown" causes. Only a series of tests, which are very simple but would require much time, could prove this assumption. Vibration should also be correlated with noises in the airplane cabin. This phase of work is fortunately well taken care of[25] and its solution is to be expected in a short time.

I desire to acknowledge my indebtedness to all Government agencies and individuals that cooperated with me in the gathering of these data and in general helped with the preparation of this and my previous paper. Special thanks are due to H. Constant, M.A., A.F.R. Ae.S., who kindly consented to being quoted at length and to the reproduction of his curves.

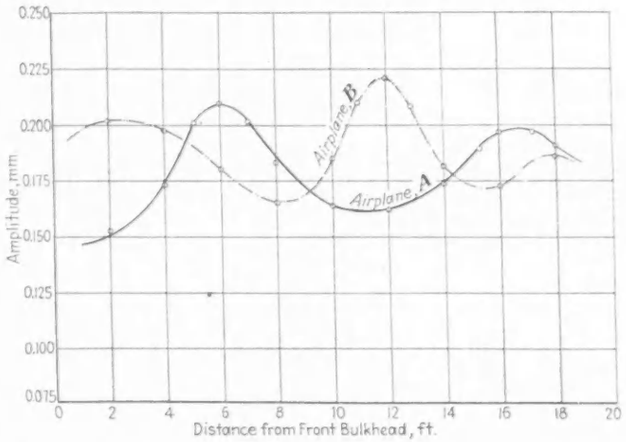

FIG. 16—CURVES SHOWING RELATION BETWEEN DISTANCE FROM BULKHEAD NO. 1 AND AMPLITUDE OF VIBRATION AS MEASURED AT DIFFERENT POINTS ON FUSELAGE TUBING IN TWO DIFFERENT AIRPLANES

[22] See A Textbook of Sound, by A. B. Wood, 1931, The Macmillan Co., New York City.
[23] See *The Journal of the Royal Aeronautical Society*, March, 1932, p. 218.
[24] See S.A.E. JOURNAL, October, 1931, p. 263.
[25] See Soundproofing of Cabins, by V. L. Chrisler and W. F. Snyder, Bureau of Standards Research Paper No. 63, May, 1929.

Refrigerator Motor-Trucks

Transportation Meeting Paper

By H. M. Williams[1] and J. W. Carl[2]

SALES potential of refrigerator trucks seems to be several hundred new units annually, according to the authors. Motor-truck transportation of meat, milk, ice cream, fruits and farm produce demands refrigeration, as does delivery-truck transportation of butter, cheese, yeast and dough. Large trucks are needed for the former class and smaller ones for the latter, but the refrigeration problems are fundamentally the same.

Desirable body-construction is outlined and the different refrigerating systems are analyzed with regard to quantity of refrigeration needed, type of insulation and insulating material available. Mechanical systems are discussed under electric, power take-off and separate gasoline-engine drives for the compressor. The most desirable location for a refrigerating unit on a truck also is considered.

REFRIGERATOR trucks are being used more extensively each year for the transportation of the perishable products of many industries. The sales potential seems to be several hundred new units annually, with the requirements of the ice-cream industry leading the list at present. The refrigerator truck is indispensable for the delivery of ice cream; therefore it is but natural that the manufacturers of ice cream have done a large amount of experimental work using all types of truck and trying practically every proposed method of refrigeration. This industry has pioneered the development and is still seeking the ideal refrigerator truck.

The transportation of meat, milk, fruits and farm produce also demands refrigeration. Trucks for these products usually are larger and utilize higher temperatures than those used for transporting ice cream. The delivery truck for butter, cheese, yeast and dough also requires refrigeration, but the truck generally is smaller. Regardless of the kind or quantity of produce being transported, the refrigeration problems are fundamentally the same. Refrigeration is required to overcome heat leakage of the truck body, losses caused by the opening of doors and by the loading of warm materials and to maintain proper temperature of the commodities being transported. The cost, weight and capacity of portable refrigerating equipment demand the most efficient body construction and insulation. Articles being transported should be cooled to the desired temperature before being moved.

Desirable Body-Construction

Refrigerator-truck-body construction requires careful consideration of the quantity and type of insulation. The best practice calls for 6 in. of cork board or its equivalent for a body temperature of 0 deg. fahr; 5 in. for 30; 4 in. for 40; 3 in. for 50; and 2 in. for 60 deg. The choice of insulating material must depend on its insulating quality, its form in connection with the body design and construction and ease of protection against moisture. Many good insulating materials are available, and their insulating factors can be obtained easily and reliably. These materials are furnished in board form and in blanket form. The boards have considerable rigidity; the blankets are easily applied but have no structural strength. Most materials utilize the well-known principle of insulation that is based on confined air cells. Aluminum foil utilizes a new principle of insulation; namely, that of reflection. It is very effective and no doubt will be used widely because of its light weight. A comparison of heat-leakage reduction is made in Fig. 1.

Regardless of the material used for insulation, it should be sealed, both inside and out, so that moisture

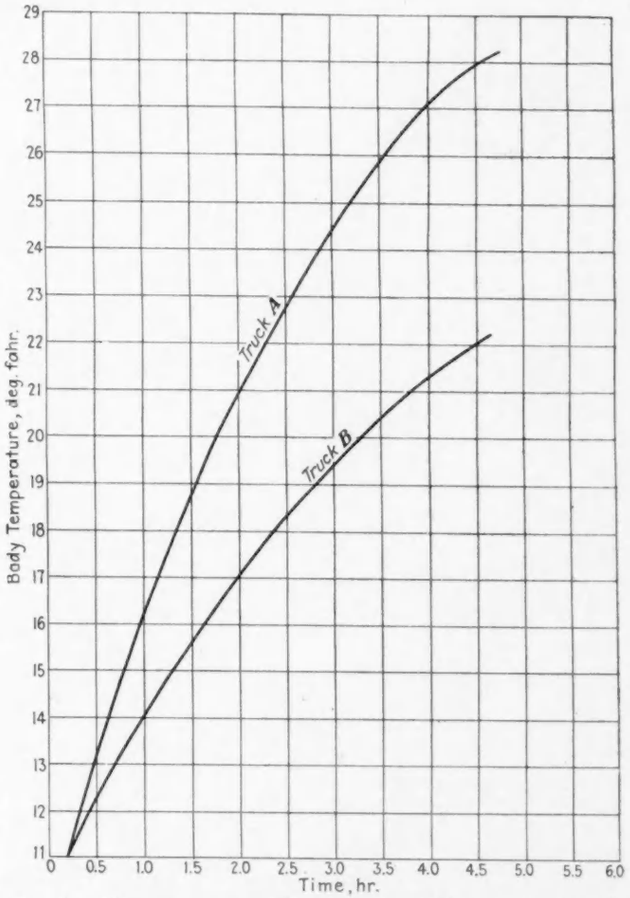

FIG. 1.—COMPARISON OF HEAT-LEAKAGE REDUCTION BY THE USE OF THICKER INSULATION

Room Temperature, 65 Deg. Fahr.

	Cork Insulation, In.	
	Truck *A*	Truck *B*
Sides	5	6
Roof	5	6
Floor	3	6

[1] Manager, commercial engineering, Frigidaire Corp., Dayton, Ohio.

[2] Commercial engineer, Frigidaire Corp., Dayton, Ohio.

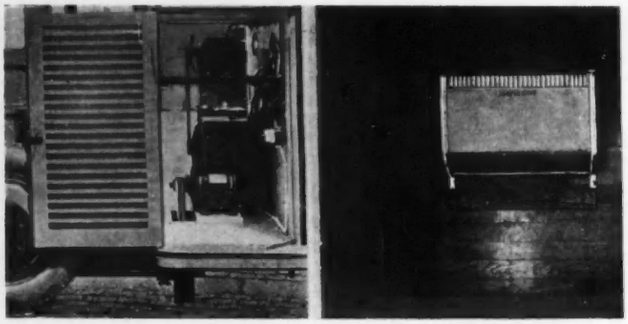

FIG. 2—(LEFT) POWER TAKE-OFF DRIVING THROUGH A GENERATOR; (RIGHT) UNIT USING DRY ICE

cannot enter and destroy its structural and insulating properties. Insulation used in the floor must be capable of supporting the load or be protected so that it will not be crushed. The floor may be of hardwood or of wood covered with metal. Where considerable moisture is present, the sealed metal covering is more desirable, as it will withstand more wear and will not retain objectionable odors. A drain should be provided in the floor to facilitate cleaning. Since cold air drops to the floor, all drains should be fitted with a liquid trap or seal.

The inside walls should be constructed of metal or of odorless lumber, as milk, butter and cheese readily absorb odors. Wood lining in meat trucks rapidly acquires the odor of smoked products. The outside walls should be of plywood, plymetal or all-metal construction. All of these materials have been used satisfactorily. The important feature throughout the body construction is to prevent any direct metallic contact between the refrigerated space and the outside atmosphere.

Allowance for the full amount of insulation should be made in building the top. If the top is crowned, the extra inch or so available should be insulated. The top is exposed to the direct rays of the sun, and temperatures as high as 140 deg. fahr. have been found on this surface. Coverings having a high reflective value are desirable.

Careful attention must be given to minimizing the number of doors and openings and to the construction of doors. Each door added means a loss of refrigeration; gaskets flatten, latches loosen, hinges sag and doors often are left open unnecessarily. Care must be used to break all metal contacts between the inside and the outside of the door by suitable door-hardware construction. Such contacts cause sweating, which results in corrosion and disintegration. The finest looking bodies are marred by rust streaks that appear around the door. The use of non-rusting metals is recommended. In the case of low-temperature bodies, provision should be made to prevent water from running into the top of the door, because this will tend to freeze it shut.

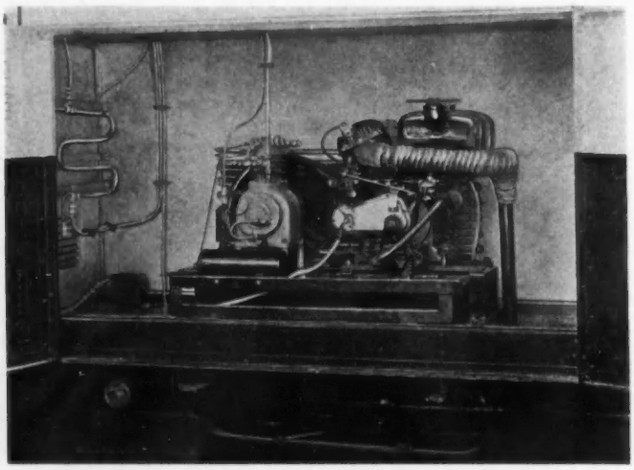

FIG. 3—SEPARATE GASOLINE-ENGINE DRIVE

Provision should be made for the mounting of ice containers or refrigerating coils at the time the body is designed. The exact type of coil hangers or supports should be known, so that their location can be planned definitely and rigid mountings assured.

Refrigerating Systems and Refrigerants

The oldest type of refrigerating system used in trucks utilizes ice, or salt and ice. The ice is either crushed or used in blocks; it may be distributed in the material refrigerated or placed in containers. Pans are often built into various parts of the body and utilize fins or other devices to increase the surface. The moisture formed is often objectionable, as it may seep into the insulation and cause deterioration. Whenever the use of salt to obtain lower temperatures is necessary, the resulting brine is very objectionable because it corrodes metal parts and shortens their life.

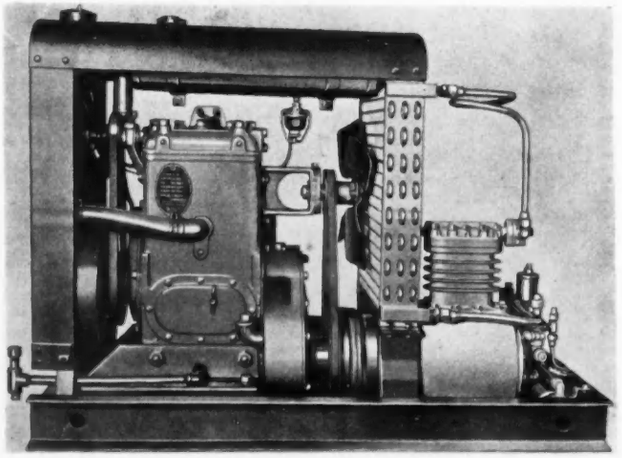

FIG. 4—DIRECT-CONNECTED GASOLINE-ENGINE DRIVE

Containers filled with eutectic solutions are used for truck refrigeration. Such systems present certain advantages and have numerous applications, although they are limited on account of the necessity for precooling and the handling charges.

More recently, solid carbon-dioxide has come into much prominence in truck refrigeration; it offers advantages over those of ice in that much lower temperatures are obtainable. It has twice the cooling value per pound and its change of form during cooling is from a solid to a gaseous state instead of being from a solid to a liquid. The fact that it produces such a low temperature makes the control of body temperatures difficult. A unit for using dry ice is shown at the right in Fig. 2.

Many types of construction have been used to distribute and control the refrigeration effect. The most recent system utilizes the double-refrigerant principle by which the solid carbon-dioxide is used to condense a secondary refrigerant such as methyl chloride or dichlorodifluoromethane. The secondary refrigerant is used to refrigerate the body and contents, its rate of cooling being controlled by suitable valves. Present prices of solid carbon-dioxide render its use rather expensive.

Some manufacturers who have large refrigerating

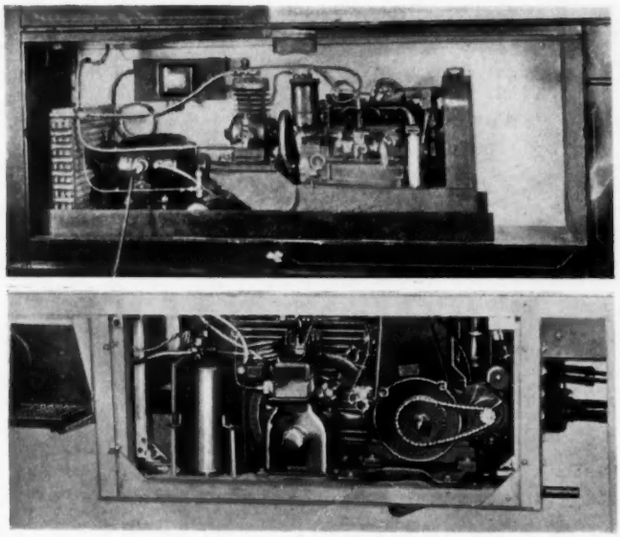

FIG. 5—(UPPER) SEPARATE GASOLINE ENGINE WITH AUXILIARY MOTOR FOR STAND-OVER OPERATION; (LOWER) FULL-AUTOMATIC GASOLINE-DRIVEN UNIT

plants use pre-cooling systems. Brine or refrigerant is circulated through the coils in the body during the time the truck is parked in the garage. The evaporator may be either exposed inside the body or imbedded in a brine tank or eutectic solution. If it is inconvenient to connect to the factory refrigerating system, a small compressor can be used. No attempt is made to control the temperature.

The foregoing methods depend on a well-constructed body and are effective for short hauls.

Many mechanical methods of refrigeration have been used. They can be classified according to the method used to drive the compressor; that is, as electric, power take-off and separate gasoline-engine drives.

FIG. 6—(UPPER) UNIT LOCATED IN FRONT, IN A WELL-VENTILATED COMPARTMENT; (LOWER) REFRIGERATOR INSTALLATION IN THE REAR

Electric Drive.—The simplest form consists in driving the compressor by a motor connected to the storage-battery of an electric truck. Storage-batteries installed on gasoline-engine-driven trucks have been used to drive the compressor. These batteries are heavy and expensive, require continuous service and attention and cause difficulty because of voltage variations.

The truck engine has been used directly to drive an electric generator for supplying energy to a motor that drives the compressor. The power-take-off method also has been used to drive the generator. Speed variation seriously affects the generator voltage and elaborate controls are necessary to overcome this difficulty. Automatic voltage-regulators usually are complicated and require expert service.

Power-Take-Off Drive.—The power-take-off method is utilized to drive the compressor direct, either through belts or gears (See Fig. 2, left). It does not permit control of the compressor speed, and this varies the capacity of the compressor. When such a variable is present, controlling the amount of refrigeration is not possible.

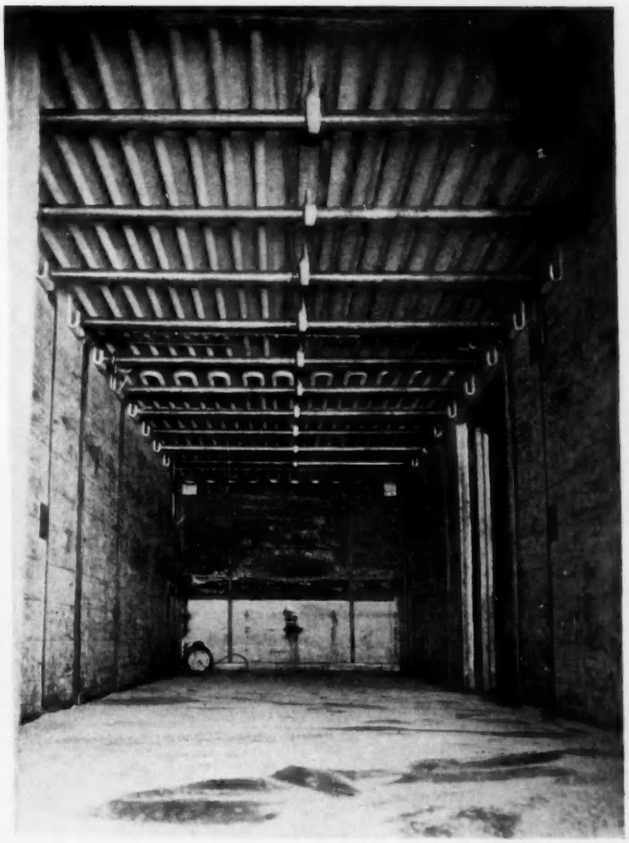

FIG. 7—METHOD OF INSTALLING PIPE COIL IN TOP OF BODY

A hydraulic device is now being developed which will deliver a constant compressor speed regardless of the driving speed. If reasonably free from requiring service, such a device should afford a satisfactory compressor drive from the power take-off. Another somewhat allied scheme consists in using a hydraulic unit. This system provides a variable-stroke pump that delivers a constant volume of oil to a fluid-propelled motor which drives the compressor; it offers possibilities but has been found too expensive and not practicable thus far.

Separate Gasoline-Engine Drive.—The unit gasoline-

engine drive incorporates a simple air-cooled or water-cooled gasoline engine for motive power (See Fig. 3). The engine can be belted or coupled directly to the compressor, making a compact refrigerating unit that is independent of any other part of the truck. One, two and four-cylinder four-cycle engines are being used. Most of these engines use magneto ignition and hand or electric starters. Although the units are heavy, they are inexpensive; the cost of operation is low and service is readily available.

Regardless of the simplicity of the engine and compressor, the combination involves problems that must be solved. Vibration causes trouble, and development work is necessary to determine a suitable type of mounting. The drive is still subject to investigation, as the power characteristics of the engine and compressor are unlike those encountered with other types of service (See Fig. 4). Close cooperation between body builders and engine and compressor manufacturers will be needed to accomplish a satisfactory solution.

Many mechanical units can be equipped with an auxiliary motor so that the compressor can be operated from a power circuit during standover periods (See Fig. 5, upper view). This motor is not essential on gasoline-driven trucks when an exhaust system is available or where the exhaust gases are not objectionable.

Automatic controls are used on many units, although the simpler type, incorporating the low-temperature cutout, seems to be more successful (See Fig. 5, lower view). The full-automatic unit has been tried again and again, but it has failed repeatedly because of its delicate control devices and the lack of understanding of its maintenance by users.

The refrigerating unit has been located in many positions on the truck. It has been placed on the top, at the front, the rear, along the sides, underneath and across the body either in the front or rear (See Fig. 6, upper and lower views). Wherever it is placed, the compartment must be large enough so that the unit is accessible. With the unit located across the body, a free circulation of air is possible. Large volumes of air are required; for example, when using a 1½-hp. air-cooled compressor for low-temperature work, approximately 15,000 cu. ft. of air per min. are necessary to dissipate the condenser heat in addition to that required to cool the engine.

Evaporators of the dry expansion type generally are used, but flooded evaporators properly located in the body are giving satisfactory service (See Fig. 7). If the evaporator is large, it should be installed when the body is being built or a false opening provided for its installation at a later date. One large section is more desirable than several small sections which may otherwise be required. Evaporators should be made of sturdy materials that will resist corrosion (See Fig. 8).

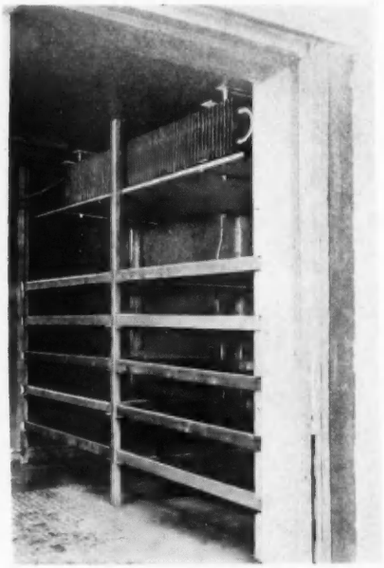

FIG. 9—UNIT-COIL INSTALLATION WITH DRIP-PAN

The evaporator must be protected by suitable baffles that will permit the free circulation of air. Refrigeration cannot be effected unless air circulation is permitted. This point is often forgotten when transporting products in boxes that can be packed close together. Suitable drain-pans should be installed under the evaporators so that drippage will not damage the products (See Fig. 9). Regular defrosting of the evaporators increases their efficiency. Ice-cream bodies are rarely defrosted, but the snow formation should be removed by frequent scraping.

Control of the refrigerant is of utmost importance. The liquid must be retained in its proper place in the system by suitable location of the expansion valve and its control. Spillage of liquid from the evaporator into the crankcase must be avoided.

We have tried to establish a few guide-posts that can be kept in mind and given consideration by truck and body builders, engine and refrigeration-unit manufacturers and users, to aid more rapid development of successful equipment for the refrigerator-truck.

[8] M.S.A.E.—Sales promotion manager, International Motor Co., Long Island City, N. Y.

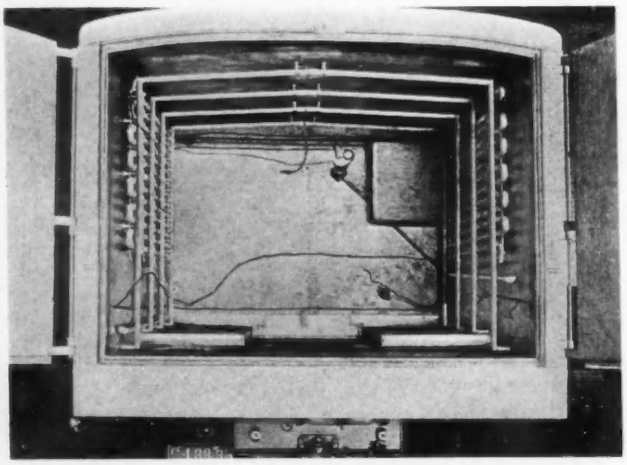

FIG. 8—METHOD OF INSTALLING PIPE COIL ALONG SIDES OF BODY

THE DISCUSSION

M. C. HORINE[9]:—Close temperature regulation of truck refrigeration is not emphasized by the authors but will demand great attention in the future. The permissible range of temperatures varies with the type of product carried; for example, fresh fish are exceedingly sensitive to small changes of temperature. According to the best information available, fresh cod and fish of similar nature should be held between 31 and 33 deg. fahr. Milk and cream can be kept perfectly fresh and usable for indefinite periods if they can be maintained within a narrow range of temperatures.

Even frozen products, such as ice cream, suffer from too wide a range of temperature. If permitted to become too warm, they melt and become unmarketable; if they are frozen too solid, they become unacceptable to the consumer and difficult for the dealer to serve. Even between these extremes, fluctuations of temperature promote the growth of ice crystals and cause a grainy texture which is most undesirable.

It is interesting to note that a system of temperature regulation has been developed which can be applied to almost any system of refrigeration and will maintain temperatures within 2 deg. fahr.

The Automotive Engine and Its Fuel

Transportation Meeting Paper

By Dr. Graham Edgar[1]

INCREASING realization by automotive engineers and by oil refiners that the nature of the fuel available largely determines engine-performance possibilities has resulted in valuable cooperative research, important progress and the preparation of a firm groundwork for future progress. The author outlines the more important characteristics that automotive-engine fuel should possess and indicates some of the relations existing between possible engine efficiency and fuel characteristics.

Heat of combustion of a fuel, volatility, tendency to detonate and freedom from impurities are discussed. Actual data from two commercial cars for which optional cylinder-heads were available are presented. These data are analyzed and the conclusions reached are that, altogether, the trend of the last few years in automotive-engine design apparently will be continued; that the fuel of the future will permit using engines of greater efficiency than those of today and that the advantages of engines of high brake mean effective pressure will be still more appreciated by the owner who drives for pleasure and by the commercial-car owner.

ONLY in recent years has an increasing realization existed among automotive engineers and oil refiners that the possibilities of the internal-combustion engine are in large measure determined by the nature of the fuel that may be available for it. This realization has resulted in a large amount of valuable cooperative research between the two industries. The further result is that important progress has been made in both industries and a firm groundwork has been laid for future progress. This paper outlines the more important characteristics that fuel for the automotive engine should possess and indicates some of the relationships existing between possible engine efficiency and fuel characteristics.

The most important properties of a fuel for the internal-combustion engine are (*a*) heat of combustion, (*b*) volatility, (*c*) freedom from objectionable impurities and (*d*) tendency to detonate.

The heat of combustion, (*a*), represents the total amount of chemical energy that is available for conversion into power by the engine. If we limit ourselves to the hydrocarbon fuels, such as petroleum products, benzol and the like, the differences between the heats of combustion, particularly when expressed as the number of British thermal units per cubic inch of engine displacement, are so small that they may be entirely neglected, and we may safely say that the heat available for conversion into power is practically independent of the nature of the fuel.

[1] M.S.A.E.—Director of research, Ethyl Gasoline Corp., Yonkers, N. Y.

The volatility of a fuel, (*b*), is of great importance in determining ease of starting the engine, smoothness of accelerating, tendency to cause vapor lock and tendency to cause crankcase-oil dilution. However, at present the volatility of practically all commercial fuels is very satisfactory from all these viewpoints and in all probability future development of the automotive engine will not be limited in any way by poor volatility of its fuel. Time would not permit a complete discussion of this subject, and it will therefore be dismissed from further consideration in this paper.

The chief impurities in gasoline, (*c*), are sulphur and gum. Here again giving time to their discussion is not worth while, since, in general, commercial fuels are very satisfactory from this viewpoint.

Tendency Toward Detonation

Ricardo, I believe, first pointed out that of all the properties of a fuel that determine its suitability for use in the internal-combustion engine, the tendency to detonate, (*d*), is so much the most important that all others can be regarded as subordinate. We have seen that the heat available for conversion into power is

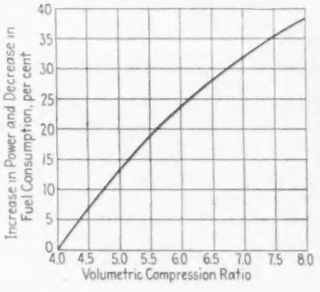

FIG. 1—VARIATION OF EFFICIENCY WITH COMPRESSION RATIO IN CHARACTERISTIC ENGINES

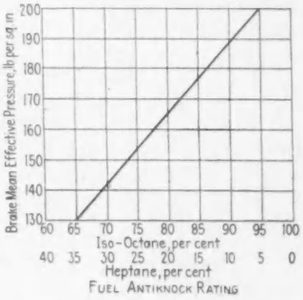

FIG. 2—RELATION OF BRAKE MEAN EFFECTIVE PRESSURE TO OCTANE NUMBER OF FUEL

essentially the same for all petroleum fuels, but the efficiency with which this heat can be converted into power depends fundamentally upon the tendency of the fuel to detonate. A brief discussion will make this clear.

The efficiency with which an engine converts the heat of combustion into power depends upon many design factors such as cylinder design, ignition and valve timing, and the like, *but in a given engine efficiency depends almost entirely upon the compression ratio;* that is, the ratio of the cylinder and cylinder-head volume when the piston is at bottom dead-center to the volume when the piston is at top dead-center. The variation of efficiency with compression ratio in characteristic engines is shown in Fig. 1.

If, however, in a given engine with a given fuel, the designer attempts to increase the efficiency by increasing the compression ratio, a point is reached where the combustion characteristics show an abrupt change; a high-pitched metallic sound occurs, the engine begins to overheat and the power, instead of increasing with increased compression ratio, begins to decrease. This

phenomenon we call detonation; or, in common parlance, "knock," "pink" or "ping." Whatever we call it, the phenomenon is of vital importance to both the automotive engineer and the oil refiner. If it did not exist, there would be no reason why brake mean effective pressures vastly in excess of those employed today should not be achieved, by increased compression ratio, supercharging, or both, with resultant increase in power per cubic inch of displacement and in fuel economy.

The nature of engine detonation has been made very clear by researches of the last few years, but the exact reasons why different fuels vary so much in their detonation tendencies still largely elude us. However, the relative tendency of fuels to detonate has been shown to depend entirely upon the chemical nature of the fuel; that is, the exact chemical structure of the thousands of chemical individuals present in the complex mixture that we call gasoline. So great are the differences produced by differences in chemical structure that at present two pure hydrocarbons differing only in their chemical structure are employed as a scale of antiknock value. One of these, pure heptane, knocks worse than any gasoline; another, iso-octane, knocks less than any gasoline. So we express the antiknock value of a gasoline in terms of the percentage of octane in a blend of heptane and octane; that is, the so-called "octane number."

No fixed relationship exists between the octane number of a fuel and the permissible compression ratio, for this depends upon engine design and characteristics; but Fig. 2, from Philip B. Taylor's paper on Increasing the Thrust Horsepower from Radial Air-Cooled Engines, illustrates the possibilities in the particular case of an aviation engine[2].

Compression-Ratio Increase Advantageous

To the automotive engineer, the advantages of being able to increase the compression ratio of engines are outstanding. With no other change than that of an increase in compression ratio, he can increase engine power (Fig. 3A); decrease exhaust-gas temperature (Fig. 3B); decrease the heat lost to the cooling water (Fig. 3C); and decrease fuel consumption (Fig. 3D). The only requirement is that non-knocking fuels shall be available.

At first sight, inspection of the diagrams shown in Fig. 3 suggests that the increase in thermal efficiency produced by moderate increases in compression ratio, while appreciable, is not great enough to affect largely the performance of the automobile. If, however, we consider the automobile as a whole, and not merely the engine, a very different picture is presented. A given automobile requires a certain amount of power to drive it on the level at a given speed; whatever additional horsepower the engine possesses is available for hill climbing, acceleration and the like. Therefore, the additional power produced by increasing the compression ratio adds a greater percentage to the reserve power than it does to the power of the engine as a whole, as shown in Fig. 5. Table 1 gives data from two commercial cars, for which optional heads are available.

TABLE 1—DATA FROM TWO COMMERCIAL CARS FOR WHICH OPTIONAL HEADS ARE AVAILABLE

	Car *A*		Car *B*	
	Low	High	Low	High
Compression Ratio	4.4	4.8	5.8	7.0
Correct Spark Advance, deg. before top dead center	10.0	10.0		¾ in. on flywheel B.T.D.C.
Increase in Average Compression Pressure, lb. per sq. in. at 20 m.p.h.		10.0		50.8
Decrease in Average Time To Accelerate from				
10 to 30 m.p.h., per cent		12.58		13.2
10 to 50 m.p.h., per cent		11.40		11.9
Decrease in Time Required To Climb				
7.2-per-cent grade (3,100 ft.), sec.		3.0		2.8
11.6-per-cent grade (1,600 ft.), sec.		12.6		3.8
Increase in Speed at Top of Grade				
7.2-per-cent grade (3,100 ft.), m.p.h.		3.5		5.0
11.6-per-cent grade (1,600 ft.), m.p.h.		10.0		5.5
Increase in Miles per Gallon, per cent at				
M.P.H.				
10				14.8
15				16.2
20		5.78		19.3
25		7.12		25.0
30		7.30		23.2
35		7.35		
40		7.70		12.0
50		8.92		11.4
60		15.38		
Increase in Top Speed, m.p.h.		2.2		3.0

It should be pointed out that the increase in ton-miles per gallon, and particularly in ton-miles of payload per gallon, is clearly greater than the increase in miles per gallon, since a substantial power increase occurs as well.

High Compression Analyzed

High compression offers to the automotive designer a means of increasing power and economy without increase in weight, with no increase in the cooling surface of the radiator and with a decrease in mean bear-

[2] See S.A.E. JOURNAL, April, 1931, p. 425, Fig. 13; or, S.A.E. TRANSACTIONS, vol. 26, p. 539, Fig. 13.

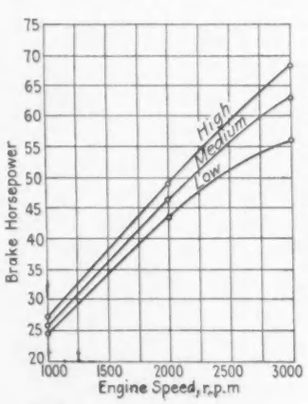

Fig. 3A—Increased Engine Power

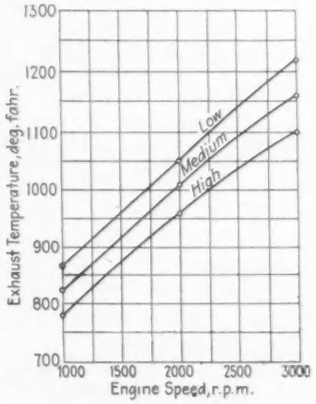

Fig. 3B—Decreased Exhaust-Gas Temperature

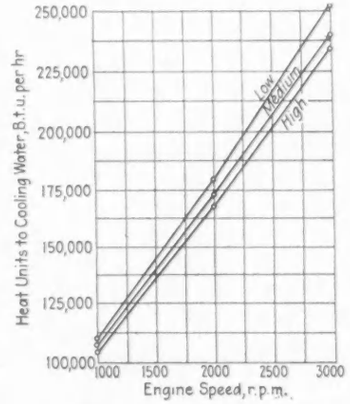

Fig. 3C—Decreased Heat Lost to the Cooling Water

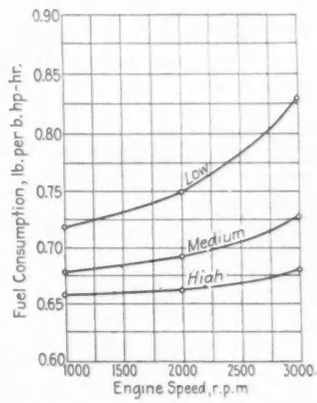

Fig. 3D—Decreased Fuel Consumption

ADVANTAGES OF INCREASING COMPRESSION RATIOS

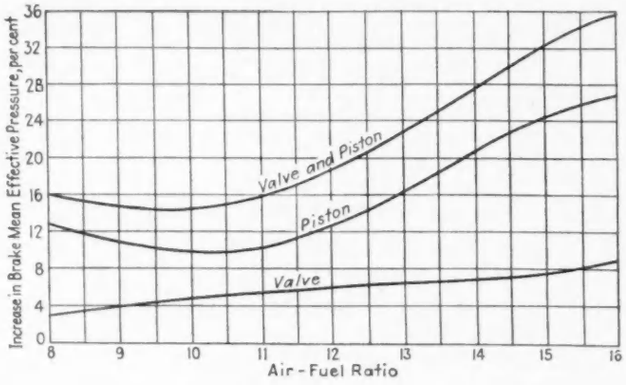

FIG. 4—EFFECT OF ABNORMAL COOLING OF PISTON AND EXHAUST VALVE ON BRAKE MEAN EFFECTIVE PRESSURE OF A THROTTLED ENGINE FOR EQUAL KNOCKING TENDENCY

ing pressures. That the average compression ratios have increased steadily in the last few years as fuels of high antiknock value have become available is readily understandable. The advantages of high compression in the field of commercial vehicles have become apparent even more recently. The increase in the power of tractors, motorcoaches and trucks as a result of increased compression has been found in many cases to more than compensate for the necessity for using high-antiknock fuels, and at present the activity among manufacturers and users in this field is marked.

While, as has been pointed out, the nature of the fuel used is the most important single factor in determining the possibilities of the automotive engine, we should not forget that the automotive-engine designer has not reached the limit of engine performance that can be attained with a given fuel.

As an illustration of some of the possibilities, Fig. 4 shows the results of some experiments in our own laboratory on the increase in permissible brake mean effective pressure *for a given fuel*, brought about by abnormal valve and piston cooling. While the devices used to cool the valves and pistons were not practical in this case, they illustrate the possibilities that the future may perhaps hold in store of making better use of the fuels at present available. The burden of future development is not entirely on the shoulders of the oil refiner; part of it must be borne by the automotive engineer. The problem is a joint one and its solution will require the closest possible cooperation between the two industries to make the most of the possibilities of the future.

Altogether, the trend of the last few years in automotive-engine design apparently will be continued, the fuel of the future will permit the use of engines of greater efficiency than those of today and the advantages of engines of high brake mean effective pressure will be still more appreciated by the owner who drives for pleasure and by the commercial-car owner.

THE DISCUSSION

M. C. HORINE[8]:—The clarity and straightforwardness of Dr. Edgar's explanations of the bearing that detonation has on the mean effective pressure which can be carried successfully and commercially in gasoline engines impress me particularly. The advantages to be derived from high mean effective pressure are so obvious as regards both operation and economy that there can be little doubt that successful means of reducing the proclivity of gasoline engines to detonate under conditions conducive to high mean effective pressure are greatly to be desired. But I am somewhat disappointed at Dr. Edgar's dismissal of the subject of volatility.

We hardly can afford to ignore any factor that offers the slightest promise of alleviating detonation. Dr. Edgar has said truly that the cooperative research carried on by the automotive and petroleum industries has resulted not only in making important progress but also in laying a firm groundwork for future progress. Therefore, I am sure that the excellence of our present fuels promises well for equally important improvements in the future. We all can agree that the gasoline of the future must be capable of operating, without knock, engines designed for higher mean effective pressures than are practical today; but whether this result will be achieved by the use of chemical brakes on combustion rates—such as tetraethyl-lead—or whether other developments perhaps as yet undreamed of hold the solution of our problem remains to be seen.

Queries on Advantages of Better Volatility

I think that we can ill afford to dismiss so important a property as volatility without careful analysis; therefore, I ask Dr. Edgar the following questions:

(1) Would not more volatility, that is, ability to vaporize at lower temperatures, permit cooler intake temperatures and therefore higher air density?
(2) Would not these things produce higher mean effective pressure, other things being equal?

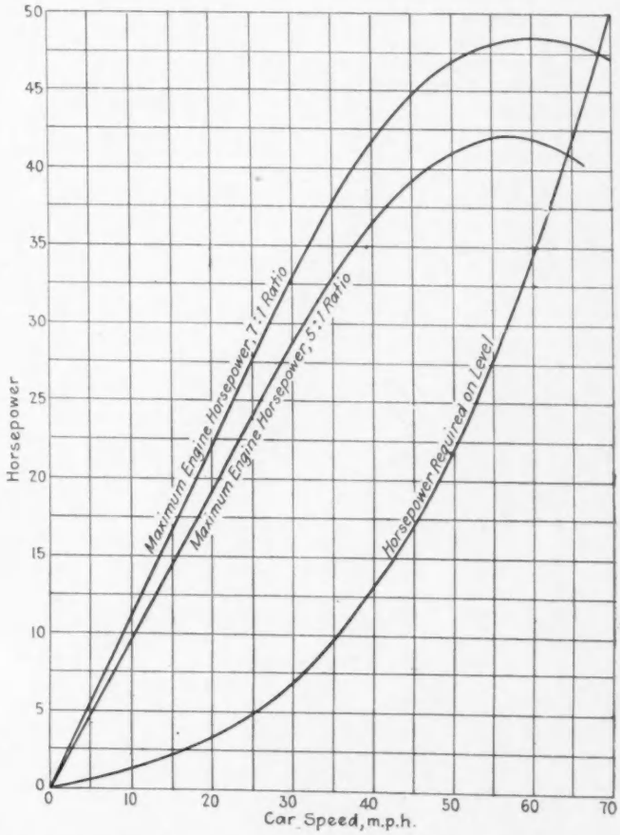

FIG. 5—POWER REQUIRED TO DRIVE A CAR AT CONSTANT SPEED, AND ENGINE POWER AVAILABLE AT COMPRESSION RATIOS OF 5:1 AND 7:1

[8] M.S.A.E.—Sales promotion manager, International Motor Co., Long Island City, N. Y.

(3) Would they not permit some increase in compression ratio without detonation?
(4) Do not lean mixtures burn more slowly than rich ones?
(5) If so, would not leaner mixtures result in less tendency to detonate?
(6) With better volatility, could we not carry leaner mixtures than we do at present?
(7) Are not prevailing mixtures usually unduly rich on account of the starting and accelerating difficulties experienced with prevailing fuels?
(8) If, therefore, fuels were more volatile, would we not use leaner mixtures, hence experience less tendency to detonate and thereby be enabled to carry somewhat higher compression?
(9) Combining the foregoing propositions, would not higher volatility enable us to use cooler and leaner mixtures and the resulting gains enable us to increase the compression successfully so as to make worthwhile gains in mean effective pressure?

The fact may very well be that, with present refining methods, and assuming that equivalent results can be achieved either way, using an antiknock dope is cheaper than increasing volatility; but this does not seem to me to warrant dismissing the possibility of better volatility, if for no other reason than that the future may hold possibilities of more economical production of high-volatility gasoline than those of which we now know.

In the infancy of the motor-vehicle, when gasoline was a by-product in the manufacture of kerosene, we used to start the low-compression T-head engines of that day "on the spark"; this despite intake manifolds made of pipe fittings, "automatic" intake valves that were opened by the engine suction, carbureters that were little more than float-feed perfume atomizers and low-tension vibrator-coil ignition. I have started such engines on the spark from 10 to 20 min. after stopping. I have seen hoar frost on the intake pipe when the exhaust pipe was fairly twinkling from incandescence. That, I maintain, was the result of volatility; it was cold carburetion. If we had such fuel to use in present engines—without hot-spot, choke or dopes—what mean effective pressures, what brake horsepowers and what miles per gallon we would get! Or would we? Like Hashimura Togo, one of Wallace Irwin's Japanese characters, "I ask to know."

Price Advantage versus Antiknock

T. C. SMITH[4]:—Dr. Edgar's interesting paper suggests to me that possibly the subjects of antiknock and detonation are covered so fully that an erroneous impression may be left to the effect that these are the only important fuel considerations. Comparing one gasoline with another, no question exists as to the desirability of having the general motoring public get satisfactory engine performance as regards antiknock characteristics. Present-day straight-run fuels have been improved in this respect to the extent that comparatively high-compression engines of good design are able to operate upon them without undesirable detonation.

Experiments that I conducted recently on a commercial passenger-car having a 6:1 compression ratio, which is considerably higher than the average, showed that with untreated gasoline having an octane rating of 72 this car could be operated practically without "pinging" on steep grades while making hard pulls. Using another untreated gasoline having an octane rating of 66, the tendency to ping was evident under some conditions; however, generally speaking, operation with this gasoline seemed to me to be satisfactory.

Judging by the large quantities of gasoline having knock ratings of from 50 to 65 that are now being sold, the operators of present-day cars seem not to be particularly worried about having a little gas knock from time to time when the use of this gasoline results in savings of a few cents per gallon of fuel.

Since Dr. Edgar has presented the other side of the problem, I trust that he will pardon a consideration that has come to my attention which favors the use of gasoline having some tendency to knock. One operator, who has used a low-priced gasoline having a knock rating of about 55, has found that the use of this gasoline has stopped the tendency of his drivers to attempt to beat all other drivers away from each stop light. They do not like to hear the engine ping, so they accelerate more slowly. His opinion is that this somewhat slower acceleration may result in less maintenance on motor-vehicles. He does not operate in hilly country.

Commercial gasolines are available that possess satisfactory volatility and freedom from impurities; but Dr. Edgar will agree, I feel sure, that with a number of gasolines now commercially available some preference exists as to volatility, particularly with regard to winter starting, and that the matter of gum and sulphur in some of the lower-priced commercial gasolines must be given consideration by the operator.

Queries on Volatility Answered

DR. GRAHAM EDGAR:—In reply to Mr. Horine, who emphasized the importance of volatility, I may point out that my rather cavalier dismissal of this subject was largely due to the limitation of time in presenting this paper.

Answering the specific questions, my opinion is that:

(1) More volatility would permit somewhat cooler intake temperatures.
(2) Cooler intake temperatures used with more volatile fuel would result in somewhat higher mean effective pressures.
(3) This would not necessarily permit any increase in compression ratio without detonation, for fuel of the same antiknock value.
(4) I do not know the relative rates of burn of lean and rich mixtures.
(5) The use of leaner mixtures than at present employed would *increase* the tendency to detonate, since the fuel-air ratio for maximum knock is distinctly on the lean side of that for maximum power.
(6) With better volatility we might be able to use somewhat leaner mixtures than those prevailing at present, particularly with a poorly designed intake manifold.
(7) The use of rich mixtures for starting and accelerating is a separate matter from the carbureter setting employed for steady operation.
(8) See item (5).
(9) I am very doubtful whether the combination of all the foregoing factors would permit any appreciable gains in mean effective pressure.

Too great volatility has certain disadvantages. Too low a 10-per-cent point increases the danger of vapor lock, increased volatility increases loss due to evaporation and too low a 90-per-cent point *may* result, with a heated intake manifold, in *less* maximum power rather than more. Altogether, increase in volatility is not the answer to the problem of increased thermal efficiency, high brake mean effective pressures and more miles per gallon.

[4] M.S.A.E.—Engineer, automotive equipment and construction apparatus, American Telephone & Telegraph Co., New York City.

Manufacturing Equipment Required To Meet the Designer's Needs

Production Meeting Paper

By L. F. Maurer[1]

I ASKED a number of designers the question, What is your greatest need in manufacturing equipment? The majority replied, Greater flexibility or versatility in tools and equipment.

Many of you are familiar with the progress made in tools and equipment with the increased demand for more and still more cars; for example, the development of special-purpose machines with fixed-center spindles, and, in many cases, very small possibility of reworking for use in a new program in case of a major change in the product. The excessive expense involved frequently delayed the making of desirable improvements even in times of very good business. How much more important it is to have equipment in which changes can be made at a reasonable cost during periods when business is not so good and management is more desirous of adopting new features for its product which will add sales appeal.

Engineering departments are being called upon as never before to refine and improve a product that already has reached a high state of development, yet at the same time management desires to restrict expenditures. Consequently, we are hearing a great deal at present as to what can be done to make changes at lower costs.

Equipment manufacturers have made a marked improvement in both standard and special-purpose machines to enable alterations to be made in much less time and at a lower cost than formerly; however, an opportunity for continued improvement still exists.

In the next five years we may see rather radical changes in the automotive industry, some of which may even threaten the existence of present manufacturing methods. Therefore very close contact between the automotive and the equipment manufacturing industries should be mutually profitable. The machine-tool builder would have time to prepare designs of equipment and the car builder would receive better machines when they were needed. In this connection, attention is called to several articles by A. F. Denham, recently published in *Automotive Industries*[2], dealing with the method of contact between the machine-tool industry and the car designer and producer. I believe that much good would come from a closer contact and interchange of ideas.

Improvements Desired by Designers

Some of the improvements in tooling, equipment and methods that are being looked for by the designer are as follows:

(1) Less costly means of changing center distances and number of spindles in multiple drilling and tapping machines
(2) Greater ease in changing centers and number of spindles without loss of accuracy and rigidity in cylinder-boring equipment
(3) Greater ease in changing over milling equipment
(4) Improved method of finish-boring cylinders to closer limits
(5) Improved methods of finishing small-diameter holes in both cast iron and steel
(6) Equipment for cheaply producing hardened-steel cylinder liners
(7) Equipment to produce high-carbon rolled-steel brake-drums more cheaply
(8) Further improvements in the production of quiet gears
(9) Improvement in concentricity of valve seats with valve guides—very vital for good performance and quiet operation
(10) Improved method of holding connecting-rod forgings to uniform weight
(11) Machining pistons and connecting-rods to uniform weight with an automatic weighing device
(12) Methods and equipment for producing welded chassis frames. This is a combined engineering, equipment and cost problem, and its solution, if found, will aid in securing frame rigidity
(13) Last but not least, ways and means of reducing the cost of body and chassis sheet-metal dies

Costs Are of Primary Importance

Great progress has been made along the lines of building dies in sections so that, in case of changes, only certain parts need altering, and, further, so that a number of different models can be made from the major parts by the use of selected sections. Nevertheless, much improvement is still desired.

Many other parts of the car in which both the engineering and the production departments desire greater accuracy and less tool and equipment expense when changes are desired could be mentioned.

You may think that I have placed too much emphasis on cost, both in changing equipment and in the resulting production, but I believe that we cannot over-emphasize cost, as our job is to give the public the greatest possible value for every dollar it spends. You may think also that to talk about making improvements is easy but to achieve them is much more difficult; however, each year some more progress is made along the lines of supplying equipment that gives the designer greater latitude, and each step helps to speed up the wheels of progress.

THE DISCUSSION

R. E. W. HARRISON[3]:—Mr. Maurer's interesting paper draws attention to a situation that faces a large number of manufacturing companies, especially those which have not followed, in the last few years, the general trend in thoroughly modernizing their equipment.

When the automobile industry first found its feet and developed into a mass-production industry, a great cry arose for single-purpose fixed-center machine-tools, and at that time it looked as though this type of equipment would be standard in the industry for all time.

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[1] M.S.A.E.—Assistant to vice-president in charge of manufacturing, Pierce-Arrow Motor Car Co., Buffalo.

[2] See *Automotive Industries*, March 12, p. 409; Aug. 13, p. 193; and Sept. 3, 1932, p. 287.

[3] Secretary, executive committee, machine-shop practice division, American Society of Mechanical Engineers, Cincinnati.

However, the development of competition within the industry resulted in the yearly-model program, which frequently entailed changes in machine-tool equipment, and this, of course, is always extremely expensive where fixed-center single-purpose machines are involved.

So embarrassing did this yearly-model change become that it completely altered the machine-tool-equipment buying-policy of the automobile industry, and for the last several years, with the exception of milling machines and some types of cylinder-boring machine, extreme flexibility has characterized the demands.

On behalf of the machine-tool industry, I can say that the tool manufacturers are always more than willing to accommodate their designs to suit a trend, and I have no doubt that Mr. Maurer and the automobile industry at large will be able to secure proper accommodation in respect of the 13 items listed in his paper. These remarks apply particularly to items (1) and (2); in fact, current advertisements in the machine-tool journals indicate that care already has been taken of this matter.

Item (3), regarding milling equipment, also has been well met, although room for considerable improvement still exists. However, the milling-machine manufacturers have latterly given great consideration to the provision of more efficient work-handling fixtures and the practice of making fixtures in duplicate so that the cutters can be kept in almost continuous engagement with the work has grown enormously.

The feeling regarding item (4) in certain quarters of the machine-tool trade is that room for some improvement as regards the design of the casting still remains. However, a considerable improvement has been made lately in the technique of boring accurate holes, as exemplified by the Heald Bore-Matic and the Ex-Cell-O machines, which utilize diamond tools and tungsten-carbide tool-bits rotating at extremely high speeds. The relatively small area of cut greatly decreases the pressure on the cylindrical bore, and an improvement in basic accuracy results.

Item (5) is a basic question, and automobile designers must realize that there always will be mechanical limitations in respect to the machining of small-diameter holes. The real and basic remedy is to design the equipment with larger holes and utilize bushings, as making a small hole accurate is relatively easy when the piece in which the hole is being made can be rotated.

In the last analysis the centerless grinding method of machining the outside diameters of these cylinder liners, item (6), is the logical way of doing this job, as this method permits the work to be ground without the distorting effect inseparable from using an arbor.

The machine-tool manufacturers have available some highly efficient brake-drum grinding machines, item (7), and my considered opinion is that real accuracy is unobtainable by other than the grinding method of finishing the working surface of the drum.

I commend to Mr. Maurer's attention in connection with item (8) the excellent paper[4] presented by F. W. Cederleaf and W. E. Sanders at a meeting of the machine-shop practice division of the American Society of Mechanical Engineers last June. This paper dealt at length with basic problems in gear manufacture and was, in my opinion, the most constructive contribution offered to the trade in recent years.

The problem in item (9) is as stated in the answer to item (5). It never will be overcome until bigger valve-guide bushings are used, thereby enabling the machine-tool equipment to function in such a way that the basic accuracy in the cylinder block is assured.

On behalf of the machine-tool builders I should like to thank Mr. Maurer for his very constructive paper. Direct labor costs have been reduced in many of the larger automobile plants to something less than 10 per cent of the total sales price of the product, largely due to better machine-tool equipment evolved as a result of cooperation between the tool user and the tool builder. As never before, the tool-builders stand ready to extend the same cooperation in the future. However, the problem is an economic one, as we cannot go on indefinitely exchanging new tools for old tools. If the improvement in manufacturing technique, as outlined by Mr. Maurer, is to be obtained, some capital investment on the part of the user will be involved. Fortunately, the precedent which would show that this investment will be a profitable one is well established.

[4] See *Metal Progress*, July, 1932, p. 19.

Material for Crankshafts

WHILE the demand for better materials is far less acute for connecting-rods and crankshafts than for other parts of the engine, a demand has arisen for harder journals. This is being met by nitralloy and by the new method of local hardening of the journals that gives greater hardness than is possible with conventional methods of heat-treatment. Nitralloy has not proved easy to heat-treat and harden in large sections, mainly because of its aluminum content, and its cost is still in the fillet mignon class whereas the entire world is clamoring for round steak and soup bones.

S.A.E.-3140 steel is in almost universal use for crankshafts of the larger heavy-duty engines to secure journals of desired hardness, although the sections required for stiffness are such that S.A.E-1045 steel provides a large factor of safety. Neither of these steels is susceptible to the degree of hardness that is desired for journals to withstand the inevitable tendency to get out of round because of the continual wear of the inner side of the crankpins at high speed.

Many engineers have observed that a car having an engine which gives excellent performance for say two years will later develop a tendency to burn out connecting-rod bearings with disagreeable frequency. This almost invariably is due to wear on the inner side of the crankpins. When the crankpin bearings are more than 0.0025 to 0.003 in. out of round, bearings are injured quickly. We have examined crankpins which showed 0.002 in. of wear after a mileage of 60,000 to 80,000, and faint marks of the original grinding would still be discernible on the outer surface which had to withstand the explosion pressure.

The solution of this difficulty may be cast-iron crankshafts with chilled shaft journals. A number of engines having such shafts are now in operation. Cast-iron crankshafts show surprisingly more fatigue resistance than steel shafts when operated to destruction with the center main bearing purposely sprung out of line 0.030 in. The average life of five steel shafts under these conditions was 16 hr., while the average of five cast-iron shafts was 62 hr. at the same speed. A number of companies have been experimenting with chilled-iron camshafts as well, and they are found to be superior to steel shafts in many respects.—From a Chicago Section paper by J. B. Fisher, M.S.A.E., chief engineer, Waukesha Motor Co.

How the Design Engineer Views Manufacturing

Discussion of Alex Taub's Production Meeting Paper[1]

R. E. W. HARRISON[2]:—I agree with Mr. Taub's ideals and sympathize with his difficulties. A careful study of his paper in the light of intimate contact as a machine-tool engineer with the automobile production departments indicates that room still remains for a much improved scheme of cooperation among the automobile designer, the machine-tool designer and the machine-tool-using superintendent.

Experience indicates that, while the machine-tool designer may have been conservative, he has seldom, or never, had the opportunity of making contact with the automobile designer except in a roundabout way through the production department. Responsibility for this lack of coordination in thought and effort reposes on the doorstep of general management or, rather, mismanagement.

No good purpose can be served by recrimination, but any experienced machine-tool engineer will readily testify to the difficulties he has experienced when trying to persuade some automobile shop superintendent to go to the engineering department with a suggested change in design that will facilitate a machining operation. He would be foolish in the extreme to try to short-circuit the shop man, as in the last analysis this is the man he must satisfy to secure payment for his product and for the continued good-will that means future orders.

The thoughtful, experienced machine-tool engineer would be delighted to meet the automobile designer in conference on a projected design, but the invitation cards hardly seem to have been printed yet, and the automobile designer is as remote as he always has been in the past. Obviously, the political economics of the situation call for action on the part of general management to get all three parties into line.

Dealing now with the two major requirements outlined by Mr. Taub.

(1) Voluntary Reduction in Variations

This is like asking a man to vote himself a 10-percent reduction in salary. Speaking as an engineer once responsible for considerable production activity, I would say "it just is not done." Control of dimensions by close tolerances always will be necessary, and the automobile designer shares with the machine-tool designer the responsibility for making it easy for the shop man to produce his work within these tolerances.

We are on the verge of a revolutionary change in manufacturing technique, particularly in the replacement of inaccurately walled castings by pressed and welded structures. The writing is on the wall for the foundryman to read the message cut out by the oxy-acetylene torch, and the scene is illuminated by the light of the many arc-welders put into service during the last several years.

(2) Increased Mobility for Change

The automobile industry originally used the then standard flexible machine-tool equipment. Mass production, with its many illusions, brought forth the demand for single-purpose machine-tools, with all their limitations and inflexibility. Competition in the trade killed the single-purpose tool, and practically all machine-tool purchases made in the last few years have covered unit-type high-power machine-tools of standard basic design and with the minimum of special equipment.

Fixed-center boring machines suffer from the limitations imposed by their virtues, and the machine-tool trade stands ready to give the automobile industry variable-center machines having ample rigidity and enough power to satisfy the shop man.

The present design of valve-tappet mechanism seems to indicate the automobile designer's lack of appreciation of certain basic factors underlying all machine-tool operations, the principle in this case being the fact that, as the diameters of machined holes diminish and their lengths increase, so the hazards of accurate machining and production cost mount up in inverse ratio. In plain language, the remedy is to provide in the engine design for relatively large-diameter holes to accommodate valve-stem and tappet bushings, thereby enabling the shop to do a good job on this component and at the same time provide the desired over-all accuracy on the assembled mechanism. While producing an accurate small-diameter hole in a big casting is difficult, this same hole can be produced with relative ease in a bushing that can be revolved and that can be ground on the outside diameter with location taken from the hole.

(3) Crankshafts

The layman is surprised to see how much any crankshaft can be sprung under light hand-pressure with the shaft carried between centers in a lathe or grinding machine, and we would be even more surprised could we see the extent to which this same shaft twists and turns and wriggles when pulling an automobile up a hill.

While poor machining will cause endless out-of-balance trouble, I have always felt that a bearing on each side of every crankpin is a necessity, not only to facilitate accurate turning and grinding, but also to hold this snaky member in place when it is subjected to the punishing use and abuse that every crankshaft is born into the world to endure.

The machine-tool designer, for his part, stands willing to guarantee any desirable limit of accuracy in the cradle-holding fixtures used for the final machining operations on automobile crankpins.

As regards accuracy of alignment of main bearings, the use of the extra-large centerless grinder, which grinds the work unrestricted by centers or clamps, seemingly is the answer. This machine, with multiple wheels, will also enable a closer check to be kept on diameter tolerances.

Advent of the lapping machine for pins and journals has gone a long way toward facilitating the production of a matured working surface with minimum potential wear, and the availability of a surface-quality calibrator enables a high standard of consistent quality to be maintained. Mr. Taub's plea for relief from the bugbear of fine splineshaft tolerances seems to have been answered in a measure by the new Barber-Coleman tapered splineshaft end.

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[1] Published in S.A.E. JOURNAL, October, 1932, p. 385. The author is development engineer with the Chevrolet Motor Co., Detroit, and a Member of the Society.

[2] Secretary, executive committee, machine-shop-practice division, American Society of Mechanical Engineers, Cincinnati.

I want to thank Mr. Taub for presenting a very interesting paper and valuable contribution. We all seem to be too departmentalized. The thought is not outrageous that, on occasion, the automobile designer should tell his shop superintendent that a certain design has been based on the assumption that it will be machined in some way prescribed by the engineer, but this situation will be acceptable only if the designing engineer is adequately posted, which is a state that I rather doubt at present.

At no time in the history of the machine-tool industry have its designing engineers been so anxious to pool their knowledge and resources with their biggest customer, actual and potential. I repeat that the liaison can be properly effected only by the general management.

Discuss Mutual Problem at Society Meetings

ALEX TAUB.—Mr. Harrison's agreement with our ambitions is appreciated and I acknowledge the lack and difficulties of contact between the machine-tool and the product designers. The main reason for this difficulty lies in the fact that between these activities lies the regular shop, which desires to, and must, control all things appertaining to production. However, we must begin. Perhaps the Society's forum and speakers' platform hold the key. The tool man can lay his developments before the engineer through the meetings of the Society just as effectively as the engineers today display their engineering progress before the membership.

Mr. Harrison objects to voluntary reduction in variations. This objection is born of the fact that he is not ready to do this. Does this mean that he never will be willing? I refuse to believe it. Improvement in method and procedure will bring this about. I am certain that, knowing our requirements, the machine-tool engineer eventually will meet them.

I do not share Mr. Harrison's optimism for the welder's torch. Castings will be with us for a long time but they will be better castings. For many reasons we should like to keep our valve-guide and stem parts small, and I am sure that the machine-tool men's ingenuity will show us how greater accuracy can be obtained in spite of the difficulties of the product design.

Crankshaft deflection has been greatly reduced in most engines, at least to a point where very little objectionable engine sensation can be credited thereto. However, out-of-balance is something else, and I am sure that Mr. Harrison can appreciate the unpleasant possibilities of a crankshaft with crankpin bearings out of position. Regardless of how well the shaft is balanced, when the large end of the connecting-rod is added to the shaft a large out-of-balance condition exists in the engine assembly. Machining the crankshaft is still very much of a machine-tool problem.

May I repeat, let us see and hear more from the machine-tool engineer at our meetings. Thus a better understanding and appreciation will grow.

Research in Engineering Colleges

INDIVIDUAL manufacturers and particularly small industries have been served rather generally by engineering colleges through special tests and investigations of their product. These have been mainly performance or service tests of a routine character that required little original scientific investigation. Until recent years, American industry in general and the larger industries in particular have given too little encouragement to research at engineering colleges. If industry desires engineering-college graduates possessing the ability to extend the frontiers of knowledge, it will be to its advantage to aid the engineering colleges in bringing their students into contact with teachers and investigators who are constantly working on the solution of new problems. While some of the larger industries may be able to solve many of their problems in their own laboratories, they must realize that they are dependent upon colleges and universities to find the material for and to develop their research staffs.

The major advantages to industry of cooperative research at carefully selected engineering colleges may be summed up as follows:

(1) The college or university atmosphere is sympathetic to research.
(2) Specialists who can furnish basic scientific knowledge are available.
(3) The institutions possess laboratory and library facilities not ordinarily found in any except the largest industries.
(4) They have means for prompt publication of results and their wide dissemination.
(5) Skilled direction of research at relatively low cost is at hand.
(6) Research usually can be conducted at educational institutions at a lower cost than is possible in an industrial laboratory.
(7) Research at colleges will assure a supply of young engineers trained to create new knowledge.
(8) The quality of engineering graduates will be improved, because students as well as instructors are brought by research into contact with new and practical problems.

Fundamental knowledge of value to the manufacturers and users of motor-vehicles has been supplied through studies at Purdue University of carbureters, pistons, steering-gears, lubricating oils, spark-plugs and automobile performance and tractor and truck tests. Considerable progress has been made in the study of the riding qualities of automobiles from the psychological and physiological as well as from the engineering standpoints.

The results secured by industry through cooperative investigations at the University of Illinois, the University of Michigan, the Massachusetts Institute of Technology, Ohio State University, the University of Wisconsin, Johns Hopkins University, Cornell University, Purdue University and several other engineering colleges indicate that such institutions can undertake and carry out successfully research projects of any magnitude. Engineering societies, trade associations, public utilities and industries both small and large will find that money invested in research at carefully selected engineering colleges will pay large dividends, not only in practical results secured, but also in increasing the supply of engineers and scientists who have the training to extend the frontiers of useful knowledge.—From an Indiana Section paper by Dean A. A. Potter, Purdue University.

The Application of Photoelasticity to the Study of Indeterminate Truss-Stresses

Aeronautic Meeting Paper

By D. L. Pellett[1]

THIS paper contains a brief description of the history, theory and application of photoelasticity, which is a new and useful optical method of stress analysis. A simple, cheap and compact photoelastic polariscope developed for the purpose is described.

The application of photoelasticity to indeterminate truss-stress analysis produces quantitative stress measurements within a maximum experimental error of 6 per cent. The tests yield with considerable speed and convenience information concerning stress concentrations at corners and fillets that cannot be obtained by other methods. The author describes a photoelastic test of an airplane-wing-rib model whereby the axial and bending stresses in the members are determined and severe stress concentrations at the spar corners are noted. The stress distribution in a truss bent, as shown by photoelasticity, is illustrated, and a design procedure that makes possible the application of the results of the tests to the design of a stamped-metal rib is developed.

MATHEMATICAL determination of stresses in an engineering structure often is an uncertain and laborious process, the degree of uncertainty depending upon the number of assumptions that must be made so that a theory can be applied to a particular case. Therefore an experimental method of stress determination that duplicates the actual conditions in the structure should be a valuable aid to engineers. Photoelasticity, which can be used to obtain the magnitude and direction of the principal stresses in any co-planar structure or co-planar element of a structure, gives us such a method.

The theory of photoelasticity is by no means new but, until very recently, it has been of interest only to physicists and many engineers are unfamiliar with the subject.

The History of Photoelasticity

The phenomenon of an accidental double refraction of polarized light was discovered by Sir David Brewster in 1815. Three years later he suggested that the double refraction of polarized light passed through a strained-glass model could be used to determine the stress in the model. However, except for occasional efforts, little progress was made along this line for almost 100 years. About 20 years ago Professors Filon and Coker, of University College, of the University of London, began research work on this problem and, with their associates, developed the photoelastic method to its present stage of usefulness.

Development of the following theories requires the use of principal stresses. Fig. 1 represents a plate of unit thickness loaded in its plane by forces *A, B, C, D, E* and *F*. Let us study the stress distribution on any infinitely small portion, such as *g, h, i* and *j*, set at angle Θ with reference axis *X-X*. These stresses can be resolved into normal components along *K-K* and *L-L*. (Fig. 2) and shear components along M_1 and M_2. The magnitude of these components will depend upon the value of angle θ_1. A value of θ_1, such as θ_2 in Fig. 3, can be found that will give a zero value of shear stress. The stresses *K* and *L* in Fig. 3, normal to planes of zero shear, are called *principal stresses*. These planes of zero shear are called *principal planes*. One of the principal stresses is the *maximum* normal stress in the material; the other is the *minimum* normal stress. If the magnitudes and directions of the principal stresses in a body are known, the shear stresses and other normal stresses can be found; that is, the magnitudes and directions of the principal stresses determine the state of stress in any body.

A beam of light is thought to be composed of an infinite number of rays having a sinusoidal vibration perpendicular to the direction of propagation. In ordinary

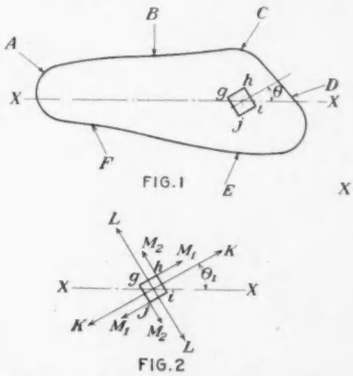


FIG. 1

FIG. 2

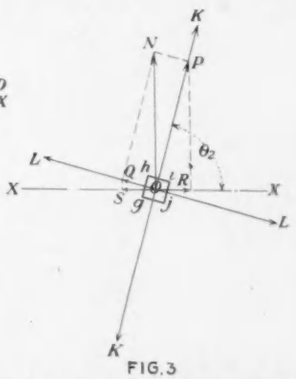


FIG. 3

Fig. 1—Plate of Unit Thickness Loaded with Forces in Its Plane

Fig. 2—Diagram of Stresses Resolved into Normal Components and Shear Components

Fig. 3—Diagram Illustrating Method of Finding Magnitude of Normal and Shear Components

EXAMPLE OF STUDY OF STRESS DISTRIBUTION IN A PLATE

[1] Formerly student at the University of Cincinnati, Cincinnati; now design engineer on special paper-making machines, Champion Coated Paper Co., Hamilton, Ohio

light, the plane of vibration of any given ray has no predetermined orientation. Plane-polarized light is composed of rays vibrating in parallel planes.

The photoelastic polariscope is composed of a light source, a polarizer, two quarter-wave plates, an analyzer, several lenses and a screen (See Figs. 4 and 5). White light from the source becomes plane-polarized in a horizontal plane after reflection from the polarizer. The first quarter-wave plate splits the polarized rays into two component rays vibrating in perpendicular planes and having a relative phase-lag of 90 deg., thus producing circularly polarized light. The second quarter-wave plate neutralizes the effect of the first plate and produces plane-polarized light vibrating in a horizontal plane. The analyzer is set so that only light vibrating in a vertical plane can pass through the prism.

The simplicity and usefulness of this polariscope seem to justify a detailed description of the apparatus. A 500-watt projection-lantern bulb is used as a light source. The filament should be as small as possible so that an approach to a point source will be effected. I have used a special lamp housing with a cooling fan mounted behind the bulb. A Nicol prism can be used as a polarizer, but I have used a stack of 12 pieces of plate glass. This type of polarizer is cheap and can be made to produce a very large beam. These plates reflect virtually 50 per cent of the incident light.

The quarter-wave plates are placed in the circuit with their optical axes mutually perpendicular and at 45 deg. to the plane of vibration of the plane-polarized beam. It is essential that these plates be true quarter-wave plates and be accurately set. Any departure from these conditions will introduce errors in quantitative results.

The analyzer is a small Nicol prism, the maximum dimension of the aperture of the prism being ¾ in. The lens sizes and spacings are somewhat arbitrary and were varied to suit the conditions in various set-ups.

The system produces a beam of polarized light about 2½ in. in diameter at the model. The model is mounted in such a position that it will be at the focal point of the third lens. It is essential that the plane of the model be normal to the direction of propagation of the light beam.

No optical effect will be produced from a transparent, isotropic sheet placed across a beam of plane-polarized light. However, if the material is stressed, double refraction occurs, and the emergent light is elliptically polarized. The directions of the lines of principal stress in the material become the directions of the planes of vibration of the component rays. As each component ray travels through the material, it is retarded by an amount proportional to the magnitude of the principal stress on the plane over which the ray travels. This change in velocity results in a relative retardation of the two wave-trains. The relative phase-lag of the component rays is given by the following equation:

$$r = (K - L)\, tC \qquad (1)$$

where

C = optical constant
K and L = principal stress values at a given point
r = relative phase-lag
t = thickness of the plate

When the stressed specimen is placed between the polarizer and the analyzer, the projected image will be covered with colored fringes. The formation of these fringes is as follows. Let ON (Fig. 3) represent the light vector of a plane-polarized ray incident upon the specimen. The specimen splits the ray into the component rays OP and OQ vibrating along the lines of principal stress, K and L. Only the vertical components of OP and OQ can pass through the analyzer. These vertical components QS and PR are vibrating in parallel planes and have a relative phase-lag, r, as previously discussed. Interference between QS and PR occurs after the light has passed through the analyzer. Certain wave lengths are damped out, while others are unaffected. This damping of certain wave lengths forms the colored fringes on the image. It will be seen that there are dark lines on the screen that travel across the image when the specimen is rotated. These dark lines, termed isoclinics, furnish valuable information in certain tests. They are the loci of all points where the direction of principal stress coincides with the plane of vibration of the incident light. Since the direction of this plane of vibration is known, the direction of the principal stresses in the isoclinics can be found. If the specimen is rotated through angular increments and the dark lines are plotted, the direction of the principal stresses at every point of the body can be found. With this information, the lines of principal stress can be plotted.

As the distribution of colored fringes depends upon the orientation of the model as well as upon the difference between the principal stresses, quantitative stress measurements become rather difficult to make. This difficulty could be overcome by rotating the polarizer and the analyzer rapidly about their longitudinal axes. Then, because of visual persistence, the color of a given spot on the image would not change as the beam is rotated. The color of the spot would then be a function of $(K-L)$ and independent of orientation of the model. Such a procedure obviously is impractical, but the same effect can be obtained by placing the quarter-wave plates in the beam of light. Circularly polarized light is equivalent to an incident beam of rotating plane-polarized light. The image produced by circularly polarized light will be covered by lines of the same color and intensity, called "isochromatics." Each of these isochromatics is the locus of all points having the same value of $(K-L)$.

Photoelastic Analysis with Models

If the stress in a strained elastic structure does not exceed the elastic limit, the stress distribution will be independent of the modulus of elasticity. This fact makes possible the study of stresses by a scale model having a modulus of elasticity that is different from that of the actual structure. Models for photoelastic work must be transparent, elastic and isotropic. Celluloid and bakelite are ordinarily used for this purpose. Glass is seldom used, as it is hard to shape and has an optical constant that is lower than that of either celluloid or bakelite. Celluloid is very easy to shape, is cheap and has an optical constant that gives visible interference at stresses much below its elastic limit. Bakelite is more difficult to handle but has compensating advantages, one of which is that the elastic limit is about 8000 lb. per sq. in., which is about four times that of celluloid. Bakelite also has an optical constant that is at least five times that of celluloid. This is a decided advantage, as the higher the optical constant is, the greater is the interference resulting from a given load.

The amount of information required is the controlling factor in a photoelastic analysis. In many cases a qualitative inspection will suffice; in other cases, accurate quantitative results are wanted. The color of the image of a model has been said to be an index of the difference between the principal stresses in the model. This stress difference can be measured easily and accurately. When the model is made, a small tension test-bar is cut from the same sheet. This bar is mounted in a dynamometer frame and placed in the beam of circularly polarized light so that the images of the bar and the model are adjacent and focused on the screen.

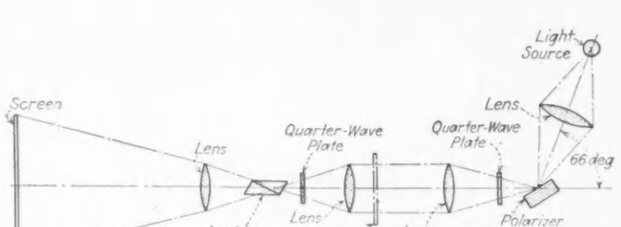


FIG. 4—SCHEMATIC DIAGRAM OF A SIMPLE PHOTOELASTIC POLARISCOPE

The load on the test bar is then increased until the image of the bar and a given spot on the image of the model are of the same color and intensity. The tension stress in the bar then will be equal to the difference between the principal stresses at a given spot on the model.

The eye is very accurate when comparing the color and intensity of two spots. However, the observation can be checked by the following procedure:

The images of the loaded test-bar and the model are superimposed and the bar is rotated until the image of the superimposed pieces has the minimum of illumination. If this image has a neutral shade, which is the shade of an unstressed specimen, the bar and the model will have the same difference between the principal stresses. This compensating effect checks the order of color in the test bar. The color of the images of the test bar and the model must be of the same order to produce a neutral shade. The angle made by the compensating test-bar and any given reference plane is the angle made by the minimum principal stress in the model and the given reference plane. This fact gives us a second method for determining the lines of principal stress.

Absolute values of the principal stresses are required in many problems but, ordinarily, we are interested only in the maximum principal stress in the structure. In most cases stress is produced by a bending moment, an axial load or a combination of both, and the maximum principal stress will occur at a free boundary of the member. Since no stress can be perpendicular to a free boundary, one of the principal stresses must have a zero magnitude. The tension in the test bar is then equal to the principal stress which is parallel to the free boundary.

Sometimes it is desirable to know the absolute principal-stress values at every spot on the specimen. These values can be obtained from the slopes of the lines of principal stress.

Application to the Study of Indeterminate Truss-Stresses

The following assumptions must be made when using the three static-equilibrium equations in the stress analysis of a framed structure:

(1) Members are connected at the joints by frictionless pins.
(2) All joints are concentric.
(3) All members are weightless.
(4) All members are straight.
(5) All loads are applied at the truss joints.
(6) The truss has no other members than those required for equilibrium.

Errors brought about by these assumptions are negligible in most problems; however, some cases occur in which these assumptions cannot be made. In these cases the stresses may become indeterminate and difficult to compute. The calculation of indeterminate secondary, or bending, stresses due to rigid connections requires the solution of a number of simultaneous equations equal to the number of rigid joints. The calculation of indeterminate stresses caused by redundant members requires the solution of a number of simultaneous equations equal to the number of redundant members.

Photoelastic analysis of a framed structure does not require the use of any of the assumptions enumerated, as all of the conditions in the completed structure can be duplicated in a scale model. As an example, I shall describe the photoelastic stress analysis of an airplane wing-rib truss. A mathematical stress analysis of this type of truss is very difficult to make and the results are questionable. Trusses of this type have curved members, the loads are not applied at the truss joints, the truss may contain redundant members, and the joints are rigid. All of these factors tend to make mathematical solutions very difficult.

Wing-Rib Test Apparatus and Procedure

A simple determinate truss, shown in Fig. 6, was analyzed and the experimental results compared with the computed results, which indicated a maximum experimental error of 6 per cent and an average experimental error of 2 per cent. Stress measurements with a tension test-bar mounted in a dynamometer frame (See Fig. 7) were made by compensation and checked by comparing the color of the test bar with that of the truss member.

Since rib designs are by no means standardized, an arbitrary design was used. The model was loaded with eight equal loads spaced according to the requirements of the Aeronautics Branch of the United States Department of Commerce, as set forth in Bulletin No. 7-A. The loading frame (Fig. 8) was arranged so that the model (Fig. 9) could be tested in positions of high angle and low angle of attack. A detailed description of these tests will be given and a design procedure developed for an actual rib.

Celluloid 0.192 in. thick was used in all tests. This material can be purchased in 20×60 in. sheets. The residual strains in them were confined to a band about 3 in. wide along all edges of the sheets. These strained edges were discarded. The elastic limit of celluloid varies from 2000 to 5000 lb. per sq. in., depending upon the age and the composition of the material. Celluloid 0.192 in. thick will have passed through the first order

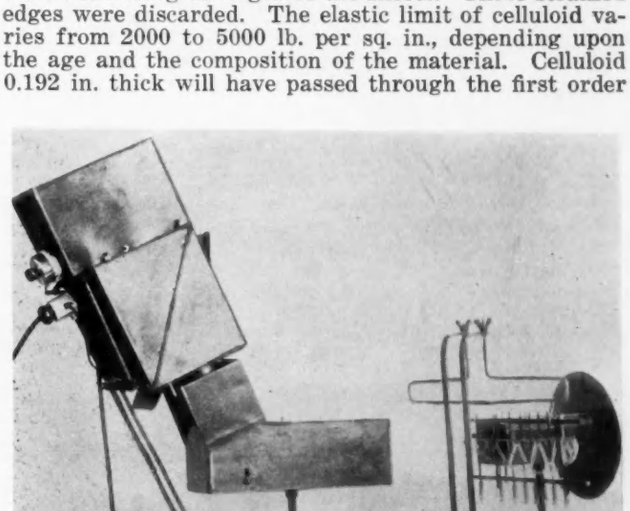


FIG. 5—TYPE OF POLARISCOPE USED IN TESTS DESCRIBED

of color when stressed to 1000 lb. per sq. in. As the colors of all parts of the strained models were kept within the first order, the stresses were kept within the elastic limit.

The knurled head of the tension dynamometer frame (Fig. 7) used in loading the celluloid test-bar is divided into angular increments of 1/20th of a revolution. The spring was calibrated in the frame with a test bar in place, thus eliminating errors due to elongation of the test bar and deflection of the frame. Stress readings were taken by counting the whole revolutions and measuring the fractional revolution on the head, one revolution of which corresponds to a stress increment of 113 lb. per sq. in. on a test bar 0.250 in. wide and 0.192 in. thick.

The wing-rib model was mounted in the test frame shown in Fig. 8. The eight loads were applied through eight springs having identical properties. Each of these springs transmitted its load through a vertical rod provided with a knurled nut. A rectangular lug was secured to each rod at the height of the chord member of the rib. The model was suspended on pivoted brass blocks corresponding to spar elements and mounted on vertical rods. Each of these rods could be moved in a vertical direction by a knurled nut. Turning both of these nuts the same amount pulls the rib down the same distance at all points on the chord. The lower chord of the rib is engaged by the lugs on the spring rods; therefore, the motion of the rib pulls down the eight rods and compresses all springs the same amount, thus applying eight equal loads to the rib. Slight corrections were made in the initial loading for the deflection of the frame and the rib. The loading frame was arranged so that it could be used to apply either high-angle-of-attack or low-angle-of-attack loading-conditions.

The polariscope used in these tests was of the type shown in Fig. 5. The beam of circularly polarized light passing through the model was about 2½ in. in diameter. The test frames were mounted in a cradle that permitted motion of the model across the beam. The model was mounted in the loading frame with load points set for high angle of attack and with all springs set for zero load. The spar blocks were raised until the spring deflection corresponded to a load of 5 lb. per spring, or a total load of 40 lb. A slight correction was then made in the spring settings to compensate for the deflections of the truss and frame. In members having severe bending, the stresses were measured at the section where the bending stresses were highest, readings being taken on both edges of the member. In members having a purely axial load, the stresses were measured at the intersection of the longitudinal and transverse center-lines of the member.

TABLE 1—STRESS MEASUREMENTS FROM CELLULOID MODEL OF A WING RIB, USING PHOTOELASTIC METHOD
(Compression Indicated by —; Tension by +)

Column	1	2	3	4	5
Member	High-Angle-of-Attack Stress Reading, Lb. per Sq. In.	Medium-Angle-of-Attack Stress Reading, Lb. per Sq. In.	Critical Stress Values, Lb. per Sq. In.	Axial Load, Lb.	Bending Moment, In-Lb.
1	+242	0	+242	—3.90	0.44
	—445		—445		
2	—138	0	—138	—5.24	
3	—168	0	—168	—6.40	
4	+312	+496	+496	+18.80	
5	+270	+410	+410	+15.60	
6	—210	—99	—210	—7.98	
7	—220	—195	—220	—8.35	
8	0	0	0	0	
9	0	0	0	0	
10	+109	+96	+109	+4.15	
11	+825	+532	+825	—2.68	1.145
	—965	—952	—965		
12	+387	+553	+553	—10.10	1.040
	—870	—1,080	—1,080		
13	+607	+520	+520	—5.40	0.847
	—675	—803	—803		
14	+430	0	+430	+2.77	0.460
	—285		—285		
15	0	0	0	0	
16	0	0	0	0	
17	0	0	0	0	
18	+330	+390	+390	+14.80	
19	—308	—354	—354	—13.50	
20	—32	—83	—83	—3.17	
21	—39	—113	—113	—4.32	
22	—240	—270	—270	—10.30	
23	—515	+546	+546	+20.80	
24	0	0	0	0	
25	0	0	0	0	
26	0	0	0	0	
27	—265	—214	—265	—10.10	

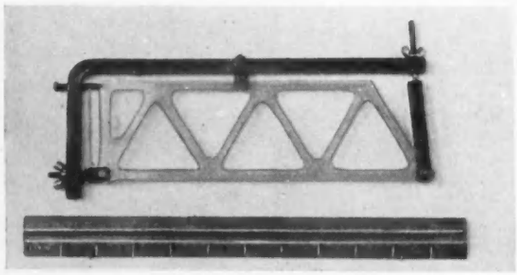
The results of these measurements are tabulated in column 1 of Table 1.

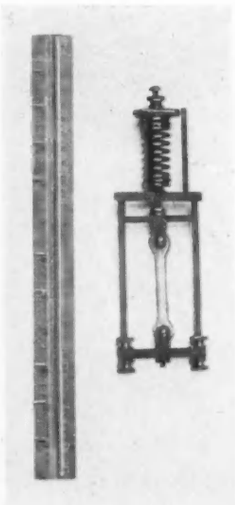
The test at medium angle of attack was conducted with a total load of 35 lb., the load factor being assumed to be 87½ per cent of the high-angle-of-attack load factor. The results of this test are tabulated in column 2 of Table 1.

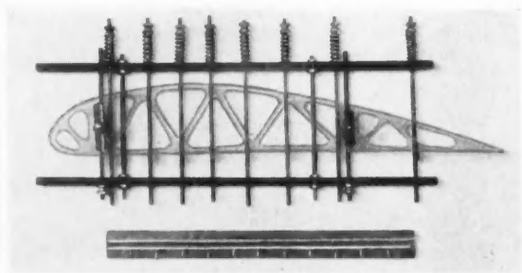
All stress measurements were taken by "compensation" and checked by placing the test bar parallel to the truss member and comparing the colors of the bar and the truss member.

Results of the Rib Test

A very pronounced stress concentration was found at the ends of those members which carried beam loads. The magnitude of these stresses for a bent composed of members 12, 20 and 21 in Fig. 9 is shown in Fig. 10. The ordinates to the curve represent the stress values at corresponding points on the inside edge of the bent. Tension is plotted toward the inside and compression toward the outside of the bent. These stress concentrations could be relieved by a substantial increase in the fillet radius. Little stress occurred in the fillets joining members carrying a purely axial load.







METHOD OF ANALYZING STRESSES IN TRUSSES BY THE POLARISCOPE

Fig. 6 (Left)—Simple Determinate Truss Mounted in Adjustable Frame for Imposing Variable Loads. Fig. 7 (Center)—Dynamometer Frame with Tension Test-Bar Mounted in It for Stress Measurement by Compensation and Comparison of Color with That of a Truss Member. Fig. 8 (Right)—Loading Frame with Celluloid Model of Wing Rib Mounted in It for Photoelastic Analysis of Stresses

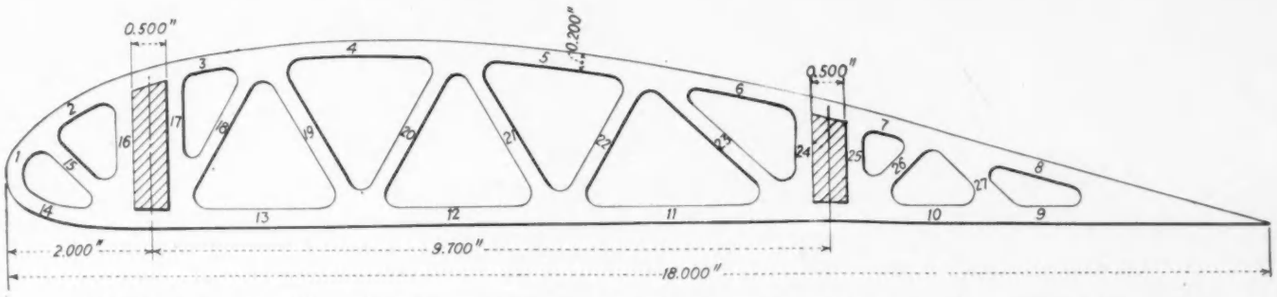


FIG. 9—CELLULOID WING-RIB MODEL ANALYZED AS AN EXAMPLE OF PHOTOELASTIC ANALYSIS OF STRESSES
This Is of Göttingen-387 Airfoil Section, All Members Being 0.192 In. Thick and 0.200 In. Wide

The most serious stress concentrations found were at the intersections of members 24 and 11 and at the spar corners of members 17 and 13 in Fig. 9. At these points the stresses were well beyond the elastic limit of celluloid. They could be relieved by the use of fillets at these points. Members 15, 16, 17, 24, 25 and 26 are redundant and were unstressed except for slight secondary stresses, which were negligible and often imperceptible in those members carrying axial loads.

The results of these tests could best be applied in the design of a stamped-metal rib, as in this type of construction the designer has great leeway in the proportioning of members and the size of fillets. The members can easily be designed if the load carried by each member of the rib truss is known, and these loads can be obtained from the information given by the photoelastic tests.

Any given member in a rib must be designed to withstand the maximum load imposed upon the member in either high or medium angle of attack. Column 3 of Table 1 contains these maximum (critical) stress values for the members in the model. The critical stress values for members in bending were resolved into axial and bending components by the following procedure:

$$S_a = (S_{t_1} + S_{t_2})/2$$
$$S_b = S_{t_1} - S_a$$
$$= S_a - S_{t_2}$$
$$S_{t_1} = S_a + S_b$$
$$S_{t_2} = S_a - S_b$$

where

S_a = axial stress in pounds per square inch
S_b = bending stress in pounds per square inch
S_{t_1} and S_{t_2} = total stresses, in pounds per square inch, taken from test readings

Column 4 in Table 1 contains absolute values of the maximum axial load in the various members. These values were obtained by multiplying the stress values in column 3, or the axial stress component in members having bending, by the sectional area of the members (0.0384 sq. in.). Column 5 of Table 1 contains the absolute values of the bending moment in the beam members. These values were obtained in multiplying S_b by the sectional modulus of the member (0.00128 in.^{3}).

The design loads and bending moments in members of any rib that is geometrically similar to the model can be found. The axial loads in such a rib can be obtained by multiplying the values in column 4 of Table 1 by the ratio between the total load on the rib and the total load on the model. The bending moments can be found by multiplying the values in column 5, by (*a*) the ratio between the total load on the rib and the total load on the model and (*b*) the ratio between the span of the rib and the span of the model. These ratios are based upon the following assumptions:

(1) Relative stress distribution in an elastic, homogeneous structure is independent of the modulus of elasticity of the material in the structure.
(2) Axial loads carried by corresponding members of geometrically similar trusses are independent of the absolute dimensions of the trusses and vary in direct proportion to the total load on the truss.
(3) Magnitudes of the bending moments in a beam vary in direct proportion to the magnitude of the applied loads.
(4) Bending moments in a beam vary in direct proportion to the length of the beam. The lengths of beam members in geometrically similar ribs vary in direct proportion to the span of the rib. Therefore, the bending moments in corresponding members of similar ribs vary in direct proportion to the span of the rib.

Due to the "theory of least work," a change in the sectional-area ratios will introduce a slight error in determining the size of a member in the rib. However, this error can be checked by a photoelastic test of the final design and can be eliminated in the second or third step of a series of successive approximations.

The photoelastic method of stress analysis can be used to obtain the magnitudes of stresses in any co-

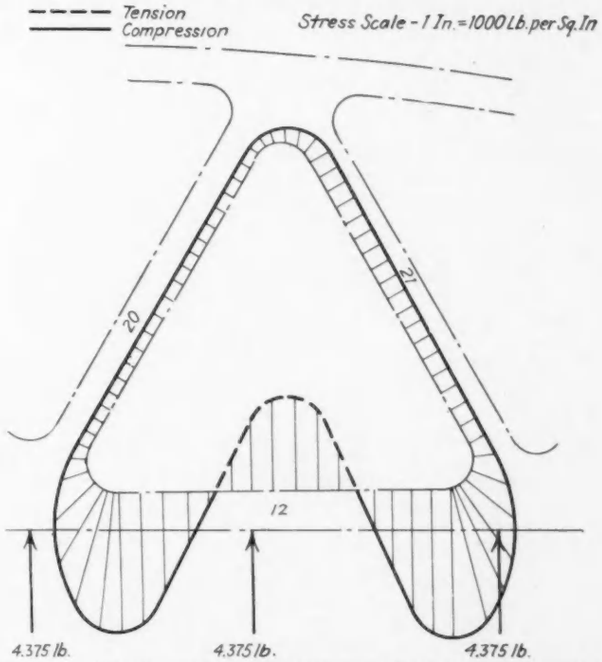


FIG. 10—MAGNITUDE OF STRESSES IN BEAM-LOAD-CARRYING MEMBERS IN A BENT OF THE RIB TRUSS
Ordinates of the Curve Represent Stress Values at Corresponding Points on Inside Edge of the Bent. Tension Is Plotted toward the Inside and Compression toward the Outside

planar system. These results are sufficiently accurate for engineering purposes, and in many cases the results will be more accurate than those obtained from a mathematical analysis. Certain problems that defy a mathematical analysis can be solved by photoelasticity.

I am indebted to the faculty of the Department of Physics of the University of Cincinnati for the interest that they have shown in the work dealt with in this paper, their valuable advice and the use of their equipment.

Simplification of Control Needed

CONSIDERATION is being given to windage and to motor-truck and motorcoach forms that will reduce it to the minimum and also provide stability. At the same time, we must give thought to the need for simplification of control to increase the certainty of operation. Unfortunately, the steps taken to simplify control have resulted, in the initial stages, in increased mechanical complication. This is evidenced by the term "gadget," which has appeared in recent months. As I see it, out of the "gadget" will emerge eventually a coordinated mechanism that will provide the desired simplicity of control and, although the mechanism in its entirety may be more complicated, this will be less obvious. It is significant, however, that the objection of complexity apparently never operates to retard the acceptance by the public of a device giving a desirable result. The functions of control afford a field in which a need for change seems evident.

With the greater loads, higher speeds and the larger mileage covered per trip or per day, it is imperative that the control of the vehicle be made as easy for the operator as possible. Large-section low-pressure pneumatic tires have made steering harder. Developments in steering-gears have been satisfactory, but it is probable that heavier front-wheel loads will be demanded, which will require still further improvement in steering apparatus. Considerable work has been done on power steering, but much more is needed before this can be regarded as acceptable. Present types give satisfactory results at slow speeds, but control at operating speeds is uncertain. A graduation of power assistance in inverse relation to vehicle speeds seems to be a necessity.

The importance of braking needs no emphasis. It probably can best be analyzed by separating consideration into two parts: operation and energy absorption. For heavy-duty vehicles, air control is gradually being developed to a point that leaves little room for criticism. The elimination of mechanical connections, the provision of means for operating trailer brakes, the equalization of pressure to the various brakes and ease of operation are all features that are most desirable. On the other hand, while great improvements have been made in brake mechanism, including drum material and friction facing, much still remains to be done. The conflict between the tire and brake to occupy the same space continues with unabated fury, and, if the extremely low-pressure tires recently proposed for passenger-cars can be accepted as an indication of what the future holds in store, the brake is out of luck. For short application under ordinary operating conditions, the present brake works satisfactorily, but, for control on down grades, the amount of energy that must be handled is too great for present mechanisms or, indeed, for any conceivable mechanism of the present type that can be incorporated within the available space. A design that will provide as nearly as possible for the dissipation of heat as fast as it is generated seems to be necessary.

The clutch and transmission system are fertile fields for speculation. The clash gear has had its full share of criticism, but no definite improvement is in sight. The gas-electric system has not received unqualified approval. Apparently, no attempt to use liquid or air as a transmission medium has progressed beyond the experimental stage. The use of the multi-range transmission is a step in the direction of a greater number of gear changes toward the ideal of infinite change, but it is open to obvious criticism. On the whole, the picture seems to be one of negation. The ideals are simple and easily stated, but the means to attain them are difficult to develop.

The ideal control should consist only of a means of steering, to be operated by hand, and one pedal lever for starting and varying the speed and another for controlling the brakes.—From a Metropolitan Section paper by B. B. Bachman, M.S.A.E., vice-president, engineering division, Autocar Co., Ardmore, Pa.

Adjusting Engineers to the Times

ENGINEERING leadership has never been in such demand as now, for the success or failure of many companies within the next two years rests on the decisions that are now being made as to exactly what shall be built and how it shall be built. At no previous time in automotive history has it been so important to analyze carefully every phase of a new design and to give the utmost attention to every detail, even details regarding which designers, draftsmen and checkers too often are inclined merely to follow precedent.

Three years ago, mediocre designs survived by virtue of an insistent demand from a market that could never be wholly satisfied with existing production facilities; today, all the broadsides of National advertising, high-pressure salesmanship and long-term financing cannot save the same designs from annihilation. Business in acceptable volume will return only to those companies that have something to offer, when a customer calls, more than a chair and a cigar.

If one thing more than another retards the improvement of a company's product, it is lack of field contact by the engineer. I do not mean merely reading reports of field representatives and branch dealers, but studying and operating equipment under the most rigorous conditions under which it can operate. Reports of service and field men, no matter how carefully written, cannot convey to the home office any adequate picture of all the factors that enter a given problem. I defy anyone to draw up a field questionnaire that will do this for any piece of equipment. If an attempt is made to cover all pertinent points, the questionnaire becomes so long and intricate that field men consider it a lot of red tape and formality, and rightfully so, for only a small proportion of the questions are likely to bear on the particular point that is being investigated.

Engineers in any industry are inclined to seek the society of friends in the same industry to the neglect of study and observation in other lines; they form their own societies, have their own meetings and specialize closely. While this has been of great value in the development of our industrial life, it has the fault of confining us to a common viewpoint for our problems, and we are likely to develop identical reactions.

Lack of proper analysis, unhampered and not influenced by a mass of conflicting opinion, sometimes retards the solution of engineering problems and the development of the product. A conference on questions of design or production may, and often does, degenerate into an endless maze of opinions, preferences, hunches and alibis, 90 per cent of which are founded on anything but facts.

Any design, no matter how meritorious, undoubtedly must take into account sales, cost and production. These problems are intimately interwoven, and it is most difficult to prevent some one of them from gaining an undue ascendancy in a discussion, so that a given line of development may be held up indefinitely. Nothing helps so much in such conferences as a carefully drawn-up list of the salient points to be covered. This should be distributed to the conferees several days in advance of the meeting, if possible, and will greatly shorten the time required to get the reactions of the various individuals reduced to definite plans of action.—From a Chicago Section paper by J. B. Fisher, M.S.A.E., chief engineer, Waukesha Motor Co.

The Economic Aspects of Transport-Airplane Design

By Ralph S. Damon[1], George A. Page, Jr.[2], and Kendall Perkins[3]

Aeronautic Meeting Paper

THIS thorough analysis of a subject that is of vital importance to airline operators reflects a great amount of experience and study. The collaborating authors endeavor to present a mathematical means for determining from cost records the type and size of airplane that will produce the largest net revenue under the operating conditions of a given route.

Conditions that influence operation are listed; factors that make up the earnings are outlined, discussed in detail and their relationship to one another and to the conditions of operation shown. An example of the earning factors on an assumed airline is set up and the effect of several airplane characteristics upon earnings are shown by operating-cost illustrations. The estimates have been kept as close as possible to the average for prospective operation.

By this presentation of a method whereby an individual operator can compare alternative types of equipment and, with his knowledge of his specific airline conditions, decide upon the type that will produce maximum earnings, the authors hope they will help to remove the selection of transport airplanes from the uncertain realm of mere opinion.

REAL dividends for commercial airlines are nearer reality now than ever before. To a large extent this may be credited to the enhanced experience and the greater competence of the operators and managers of the present airlines and particularly to the rapidly growing habit of watching and analyzing the figures that the accounting department, after many months of seemingly purposeless effort, produces and often too humbly presents to the management. These figures, when studied understandingly, tell a rather complete story of why one airplane fails to pay for itself while another more than does so; why one airline loses money while a similar one thrives; why last year's decision on purchases determined the difference between a profit and a loss. In short, they furnish a simple answer to the question: How well has the organization done its job? The very life of both operator and airplane manufacturer rests upon this answer, and the fact that both are giving it the most thorough consideration is not surprising.

Probably the most serious thought has been given to the type of flying equipment, since this is at the very foundation of the expense picture and represents by far the most fertile field for growing economy. Some of the prominent operators have expressed the results of their study of this problem, and no doubt many others have left their thoughts unexpressed. The wide variation in the conclusions reached about a subject that seems so purely a matter of accounting is striking and at first a little surprising. In each case the arguments seem unanswerable, but in all probability a point of reasonable compromise exists somewhere, which, as in the case of most complex problems, comes nearer the optimum economy than either extreme. The problem deserves further discussion before coming up for vote.

Expenses and costs must be studied as a group rather than as individual items. A cost accountant would no more estimate the cost of operating an airplane on the basis of his fuel mileage or his first cost alone than he would determine the cost of manufacture on the basis of labor time alone. After all, the fact that usually is vital is, not what the tires on the new airplane cost, but how much will the earnings be in the next several years?

The purpose of this discussion is to help clarify the analysis of costs so as to enable an operator to determine for himself the flying equipment that, together with his particular route conditions, will lead to the maximum figure for net earnings.

Several methods of approach are available. The one used will be (*a*) to list the various conditions that have an influence on airline operation; (*b*) outline the factors that go to make up the earnings of an airline; (*c*) discuss these factors in detail and show their relationship with one another and with the conditions of operation; (*d*) set up an example of the earning factors on an assumed airline and (*e*) show by illustration the effect, measured in earnings, of several airplane characteristics.

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[2] M.S.A.E.—Chief engineer, Curtiss-Wright Airplane Co., St. Louis.
[3] Service engineer, Curtiss-Wright Airplane Co., St. Louis.

We shall begin with a list of the conditions that affect earnings. The variety of such conditions is so great that for clarity we shall tabulate first the various categories into which they will fall, as follows:

CLASSIFICATION I—BY EFFECT ON EARNINGS

(1) Operating expense
(2) Fixed, semi-fixed and independent charges
(3) Revenue

CLASSIFICATION II—BY VARIABLE CONDITIONS

(1) Conditions controlled by type of flying equipment (called airplane variables)
(2) Conditions controlled by operating conditions (called operating variables)
(3) Conditions controlled by organization efficiency, other than in selection of flying equipment (called efficiency variables)

CLASSIFICATION III—BY APPLICATION

(1) Conditions that affect costs directly (called direct variables)
(2) Conditions that affect costs indirectly (called indirect variables)

Application of Variable Conditions

We can now study the application of the variable conditions in their effect on earnings.

The economics of transport operation is both a science and an art. It is a science to the extent that the effect of the various conditions can be measured and studied arithmetically, and it is an art to the extent that the effect of those conditions which cannot be measured must be judged on the basis of experience and common sense. Thanks to improving statistical information, the number of measurable conditions is increasing, although there still are many, and always will be some, of those that cannot be expressed numerically. To present a complete picture, the unmeasurables will be listed, although their weight in determining operating and purchasing policies must be left entirely to the good judgment of the operator. Nearly all of the indirect variables are in this category.

OPERATING EXPENSE

(1) Direct airplane variables
  (*a*) Cruising speed
  (*b*) Payload—passengers, baggage, mail, express
  (*c*) Initial cost of airplane with complete equipment. Scrap value of airplane and complete equipment
  (*d*) Initial cost of engine(s). Scrap value of engine(s)
  (*e*) Life of airplane
  (*f*) Life of engine(s)
  (*g*) Fuel consumption at cruising speed
  (*h*) Oil consumption at cruising speed
  (*i*) Spare parts and material for airplane maintenance, repair and overhaul
  (*j*) Spare parts and material for engine maintenance, repair and overhaul
  (*k*) Standard number of hours of labor for airplane maintenance, repair and overhaul
  (*l*) Standard number of hours of labor for engine maintenance, repair and overhaul
  (*m*) Insurance or reserve for fire, crash, theft, windstorm and so forth (air risk)
  (*n*) Insurance or reserve for public liability, passenger liability, property damage, accidental damage and so forth
  (*o*) Insurance for crew compensation
  (*p*) Number in crew
  (*q*) Ratio of runs completed to runs scheduled (due to airplane)
(2) Indirect airplane variables
  (*a*) Landing speed (under operating conditions)
  (*b*) Climbing or gliding angle with one engine dead (under operating conditions)
  (*c*) Take-off run and climbing angle (under operating conditions)
  (*d*) Use of co-pilot
  (*e*) Installation of safety and economy equipment
  (*f*) Pilot's visibility
  (*g*) Ease and effectiveness of control
  (*h*) Stability
  (*i*) Structural and mechanical integrity
(3) Direct operating variables
  (*a*) Price of fuel
  (*b*) Price of oil
  (*c*) Weather conditions (permissible frequency)
  (*d*) Traffic density
  (*e*) Distance between stops (speed)
  (*f*) Altitude (landing and flying)
  (*g*) Rate of pay for crew
  (*h*) Rate of pay for mechanics doing maintenance, repair and overhaul
  (*i*) Inspection
  (*j*) Compensation insurance (crew)
  (*k*) Ratio of runs completed to runs scheduled (due to operating conditions)
  (*l*) Number of route miles in system
(4) Indirect operating variables
  (*a*) Weather conditions (safety)
  (*b*) Terrain conditions
  (*c*) Landing-field conditions (size, altitude, surface conditions and obstructions)
  (*d*) Number of emergency fields
  (*e*) Distance between stops (safety)
  (*f*) Airway marking and lighting
  (*g*) Competition
(5) Direct efficiency variables
  (*a*) Ratio of actual labor hours to standard labor hours for maintenance, repair and overhaul
  (*b*) Replacement of equipment when it is most economical to do so, regardless of age of that in use
  (*c*) Time lost at stops
  (*d*) Percentage of time airplanes are in operation
  (*e*) Intelligent purchasing
  (*f*) Creation and maintenance of maximum traffic density
  (*g*) Wise management in handling numerous details
(6) Indirect efficiency variables
  (*a*) Pilot's skill
  (*b*) Safety policies
  (*c*) Morale

FIXED, SEMI-FIXED AND INDEPENDENT CHARGES

(1) Administrative overhead
(2) Supervision overhead
(3) Accounting overhead
(4) Traffic management
(5) Advertising and traffic promotion
(6) Printing
(7) Taxes
(8) Rents
(9) Capital charges on fixed assets other than flying equipment
  (*a*) Hangars
  (*b*) Stations
  (*c*) Offices
  (*d*) Shops
  (*e*) Ground equipment
  (*f*) Furniture and fixtures
  (*g*) Shop supplies and inventory
  (*h*) Office and station supplies and inventory
  (*i*) Radio and special landing equipment
  (*j*) Land
  (*k*) Miscellaneous
(10) Ground transportation
(11) Clearance and customs
(12) Personnel equipment
(13) Heat, light and power
(14) Janitor and watchman service

(15) All labor not applying directly to airplane or engine maintenance, repair or overhaul
(16) Purchasing
(17) Information and stenographic service
(18) Engineering
(19) Stores
(20) Expenses due to omitted or uncompleted trips (other than loss of revenue)
(21) Legal expenses
(22) Operations management
(23) Porter service
(24) Radio operation
(25) Shop supplies
(26) Office supplies
(27) Communications
(28) Insurance on assets other than flying equipment, fire, windstorm, hail and theft
(29) Maintenance and repairs for assets other than flying equipment
(30) Traveling expenses
(31) Miscellaneous expenses

REVENUE

(1) Direct variables
  (*a*) Fare
  (*b*) Passenger traffic
  (*c*) Excess-baggage traffic
  (*d*) Mail traffic
  (*e*) Express traffic
  (*f*) Percentage of capacity
  (*g*) Income from subsidiaries
  (*h*) Special operations and services
(2) Indirect variables
  (*a*) Passenger good-will (repeat business)
    (1) Passenger comfort—noise, vibration, seats, space and so forth
    (2) Destination speed
    (3) Personal service
    (4) Advertising and traffic promotion
    (5) Schedule frequency and infallibility
    (6) Appearance of equipment and personnel
  (*b*) Public good-will (new business)
    (1) Publicity
    (2) Advertising and traffic promotion
    (3) Passenger good-will
    (4) Schedule frequency
    (5) Appearance of equipment and personnel

How Conditions Affect Earning Factors

We can now investigate the factors that make up the final earnings. Discussion of each of these factors in detail will be necessary to show how it is affected by the conditions that have been listed. The final results will be based on total number of dollars earned per year, because this will best show the effect of various conditions on the size of earnings. For the sake of simplicity, the revenue derived from carrying mail, express and excess baggage will be omitted from the calculations. Most of the large airline operators receive a substantial portion of their revenue from mail operation, but the basis on which contracts are awarded and on which payments are made and the knowledge of what future mail contracts will be are very indefinite. An individual operator knows what his particular mail possibilities are and can easily add the revenue and the expenses involved to those for passenger operations. Express and excess baggage can be treated similarly without undue complication.

The fixed, semi-fixed and independent charges on different lines vary with the operating conditions and the accounting set-up. When the division of accounts required by the Post Office Department is used, the direct operating costs, plus insurance that can be charged to operations, average roughly 60 per cent of the total expense. This leaves about 40 per cent of the total for the other charges. They have, however, little bearing on the type of flying equipment to be used and will also be omitted from the estimates. For purposes of illustration only, the assumption, which seems logical, will be made that these charges will be balanced by the revenue from mail, express and like services.

Factors Expressed by Equations

The general factors that affect earnings from passenger operation can be expressed by several simple equations:

Revenue from passenger operation (dollars per year) = total airplane miles per year (for all airplanes) × passenger fare (dollars per passenger per mile) × average number of passengers per airplane (1)

Operating expense (dollars per year) = total miles per year (for all airplanes) × operating costs (dollars per hour) ÷ average speed (m.p.h.) (2)

Earnings from passenger operation (dollars per year) = revenue from passenger operations − net operating expense − capital or interest charges on flying-equipment investment (3)

Revenue from Passenger Operations

Let us now discuss these factors in detail. All the *direct* factors that make up this revenue are determined largely by the conditions of operation and by the management of the airline and do not directly depend upon the type of flying equipment. The large majority of passengers, present and prospective, do not care whether the machine in which they fly is made of steel or wood, or whether it carries 500 or 5000 lb. of payload.

Many of the *indirect* factors of revenue are determined by the airplane but cannot well be measured. The transport business, catering as it necessarily does to the most comfort-accustomed portion of the traveling public, must necessarily provide transportation that is superior in speed and at least equal in other respects to all other means of travel. If provision is not made for noise and vibration reduction, ample space to move around in and seats that are comfortable and roomy, and if the personnel or equipment does not look capable of performing its job efficiently, the result is a loss of patronage which may have a disastrous effect upon net earnings. In our calculations it will be possible only to estimate roughly on the basis of railroad experience that the value to the passenger will be 5 per cent more for travel in an airplane that is fast and well equipped than in one that has 85 per cent of the average speed and less comfort. Future fares per passenger-mile on this basis might be $0.050 and $0.045 respectively.

Operating Expense.—The factors that make up operating expenses will be discussed individually.

Total Miles per Year.—The number of miles flown per year is decided by the operating conditions and the number and speed of airplanes. It can be expressed in the form of an equation:

Total miles per year = miles per run × number of airplanes × schedule runs per airplane per year × ratio of runs completed to runs scheduled (4)

This same factor, of course, enters into the formula for revenue.

The number of route miles is definitely fixed by operating conditions. The number of airplanes and the number of schedule runs can be decided by a cost analysis made on a basis similar to that to be shown hereafter, together with an analysis of the particular conditions surrounding the route. The approximate ratio of runs completed to runs scheduled throughout the year also can be approximately determined in the latter analysis.

Operating Costs per Hour.—These costs are divided somewhat differently on the various airlines, but the

method of division is not important to the principle. The following classification, in which all costs are in dollars per hour of actual flight and taxying, might be used:

Depreciation on airplane, less engine(s)
Depreciation on engine(s)
Gasoline and oil
Spare parts, materials and labor for airplane maintenance, repair and overhaul
Spare parts, materials and labor for engine maintenance, repair and overhaul
Insurance or reserve for fire, crash, theft, windstorm and so forth
Insurance or reserve for public liability, passenger liability, property damage, accidental damage and so forth
Pay and compensation insurance for crew

The sum of all these costs may be called the net operating cost per hour, as it includes no interest or capital charges on the investment in flying equipment.

The various items will be discussed individually, as follows:

Depreciation on Airplane and Engine(s).—Depreciation of an airplane, like that of nearly all machinery, is on an arbitrary amount set aside for replacement when the economic utility of the plane has vanished. It is based on obsolescence, inadequacy, supersedence or decrepitude. At present the basis for replacing airplane equipment probably should be economic obsolescence, and that for engines, decrepitude, although any one of the other factors may enter. Upon retirement, unless converted to use as freighters, the scrap value of the airplane may, for purposes of conservative accounting, be neglected. This value is estimated on the basis of recent operation experience that the economic life of present airplanes is 5000 hr. and that of engines 2500 hr. The charge per hour for depreciation on one unit will therefore be:

$$\text{Depreciation on airplane} = 0.0002 \times \text{initial cost of airplane} \quad (5)$$

$$\text{Depreciation on engine(s)} = 0.0004 \times \text{initial cost of engine(s)} \quad (6)$$

Gasoline and Oil.—The consumption of gasoline and oil for engines of various types is found to be almost exactly proportional to power. The fuel consumption per hour for the over-all engine operation, at cruising speed, is approximately 6.0 gal. per 100 hp.; so, if we assume an average price of $0.15 per gal., the cost will be $0.060 \times \$0.15 = \$0.009$ per hr. per hp. The consumption of oil per hour, including replacement, can be expected to be roughly 0.4 gal. per 100 actual hp., and if we assume the average price paid to be $0.70 per gal. the cost will be $0.004 \times \$0.70 = \$0.002$ per hr. per hp. The charges for gasoline and oil per hour can be combined and expressed in the form:

$$\text{Gasoline and oil} = 0.011 \times \text{maximum actual horsepower} \quad (7)$$

Spare Parts, Materials and Labor for Maintenance, Repair and Overhaul of Airplane.—The costs for these items are a large proportion of operating expense but are found to differ widely as among the various airlines. Possibly differing accounting setups or shop efficiencies explain some of the difference, but the greatest portion can be traced directly to the type of flying equipment. The costs depend primarily upon the following factors: (*a*) size, (*b*) type of construction, (*c*) suitability of design, and (*d*) care in handling.

Size of the airplane really affects only the cost per airplane. Variation in cost per passenger seat per hour due to size alone is insignificant and can be neglected.

Effect of Type of Construction on Maintenance Cost

Type of construction, on the other hand, undoubtedly is the largest factor determining maintenance costs. One airline operating all-metal machines almost exclusively sets aside, as reserve for all maintenance, repair and overhaul per hour, approximately 0.00016 times the initial cost of the airplane. Another, using exclusively airplanes with metal-tubing construction and fabric covering, estimates these costs to be roughly 0.00007 times the initial price. A seemingly very conservative assumption is that, in the present stage of design experience, the cost of maintenance, repair and overhaul will be at least 40 per cent greater for an all-metal airplane than for one of steel-tube construction and fabric covering. The cost of overhaul alone probably is approximately 100 per cent greater for the all-metal type. In any case the individual operator can obtain more exact figures to cover his particular conditions and can determine for himself the price he pays for metal construction.

Let us set up two formulas, one for metal construction and one for all other types. Reasonable figures for the cost per hour for spare parts, materials and labor for maintenance, repair and overhaul would be, approximately,

$$\text{For all-metal airplanes} = 0.000115 \times \text{initial cost of airplane} - \text{engine(s)} \quad (8)$$

$$\text{For other types} = 0.000095 \times \text{initial cost of airplane} - \text{engine(s)} \quad (9)$$

These factors would show considerably more difference if the initial cost of the airplane in each case were the same.

Spare Parts, Materials and Labor for Maintenance, Repair and Overhaul of Engines.—Charges for these items are based upon conditions corresponding to those for the airplane. Many of the detail costs depend roughly on the total number of actual horsepower, but others depend on the number of engines. For purposes of simplicity and until more information is available, we shall assume that the total charges for spare parts, material and labor depend upon the total number of actual horsepower.

Horsepower is chosen rather than first cost because the relationship seems more reasonable. Engines in which the horsepower is increased by supercharger and compression-ratio increase show a maintenance cost that is larger in proportion to original cost than in the case of standard engines. Also, engines manufactured cheaply in the first place show a larger figure for maintenance. A reasonable average figure per hour for these charges is 0.0055 × total actual horsepower.

This figure does not give an exactly true picture of engine costs, because the cost for labor to maintain three 400-hp. engines has been found to be considerably more than for two 600-hp. engines. The exact difference will have to be determined by comparative experience.

Insurance or Reserve for Fire, Crash, Theft, Windstorm and So Forth.—Airplane insurance is still largely a matter of estimate, the rate for any given machine over any particular route being fixed arbitrarily at first and then adjusted to agree with experience as time goes on. It depends on original cost; years, hours and miles operated; passengers carried and similar factors. We shall assume that the basis is a multiple of the initial cost and number of hours operated. Insurance can be purchased in any one of several ways, perhaps the two most common being

(1) Coverage for nearly the full initial cost during the entire life of the machine
(2) Coverage for the full depreciated value of the machine during its economic life

Assuming that the second method is used, the rate might be (per hour for both airplane and engine)

$$\text{Fire, crash, theft and windstorm insurance} = 0.000058 \times \text{initial cost of complete airplane} \quad (10)$$

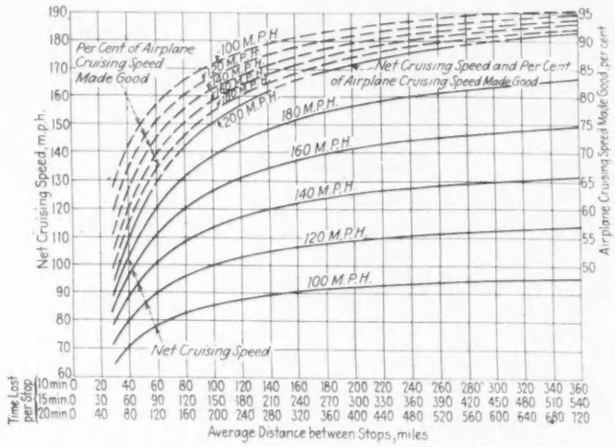

FIG. 1—RELATIONSHIP BETWEEN AIRPLANE NET CRUISING SPEED AND AVERAGE BLOCK-TO-BLOCK SPEED

Curves Developed by E. R. Child, Based on Average Miles between Stops and Average Time Lost at Stops

Insurance or Reserve for Public Liability, Passenger Liability, Property Damage, Accidental Damage and So Forth.—We can assume as a reasonable figure per hour for these charges 0.000065 × initial cost of complete airplane.

Pay and Compensation Insurance for Crew.—These costs have been fairly well established and usually approximate $7.50 per hr. for a pilot and $10.00 per hr. for a pilot and co-pilot.

Speed as a Cost Factor

Average Speed.—By average speed, or "block-to-block" speed, is meant the number of miles from the airport at which a run originates to the airport at which it terminates, measured along the flight path, divided by the time between leaving the blocks at the originating airport and the time of arrival at the blocks at the terminal airport. The relationship between the average and flying speeds is interesting. Fig. 1 shows curves developed by E. R. Child, of Buffalo, to show this relationship on the basis of average number of miles between stops and average time lost at stops. The time lost includes all taxying, gliding, climbing and waiting time; and the amount of such time makes a remarkable difference in the average speed. For example, 10 min. lost in 200 miles = 15 min. lost in 300 miles = 20 min. lost in 400 miles; and, from the chart, under one of the foregoing conditions, we find the data presented in Table 1.

The distance between stops is governed by operating requirements, but the time lost at each, while affected by the size and performance of the airplane, is largely a matter of efficiency in handling and loading the machine on the ground. From the foregoing tabular data and the fact that all operating costs except capital charges on flying equipment depend upon average speed, the extreme importance of short stops becomes evident. The railroads have a similar problem and are noted for the expedition with which they handle stops. Because of the greater speed on airlines, the importance of this is at least twice as great.

TABLE 1—RELATION BETWEEN CRUISING AND AVERAGE SPEED

Cruising Speed, M.P.H.	Average Speed, M.P.H.	Per Cent
120	109	91
140	125	89½
150	133	89
160	141	88
180	156	87

Cross wind is another factor in speed correction, but this is small enough to be neglected in our calculations.

Cruising Speed.—The cruising speed of an airplane of a given weight, at a fixed acceptable number of engine revolutions per minute, depends primarily upon three airplane conditions

(1) Aerodynamic efficiency
(2) Horsepower
(3) Wing area

The relationship between these can be expressed roughly as follows:

$$\text{Speed} = \text{aerodynamic efficiency} \times \sqrt[3]{(\text{horsepower/wing area})} \quad (11)$$

Aerodynamic efficiency has been expressed by an arbitrary factor for comparison between various designs. It is found to vary, on an airplane of given weight, with size, shape and wing area, the size usually being less important than the shape. These factors will be discussed separately. The aerodynamic efficiency of the flying equipment now shows up as an all-important factor in determining speed and hence operating costs. Prediction can be made that as time goes on this item will receive more and more attention.

Calculation of Economical Horsepower

Horsepower.—The speed of an airplane, other conditions being equal, varies approximately with the cube root of the actual horsepower of the engine or engines at the operating speed. Thus, if the actual horsepower is doubled in a particular machine, the speed can be expected to increase, roughly, 26 per cent.

The power that can be obtained practically from an engine of a fixed displacement has certain limits. On the basis of this maximum actual power and the net sales prices as of April 1, 1932, the price per horsepower of 17 standard engines suitable for transport use was found. The average of these was $10.60 net per actual

TABLE 2—AERODYNAMIC-EFFICIENCY FACTORS OF 10 COMMERCIAL AIR TRANSPORTS

No.	Number of Engines	Wing Arrangement	Approximate Actual Payload, lb.	Approximate Total Maximum Actual Horsepower	Approximate Wing Area, Sq. Ft.	Approximate Maximum Actual Speed, M.P.H.	Aerodynamic-Efficiency Factor
1	One	Monoplane	1,800	605	418	140[a]	124
2			1,100	655	314	165	129
3			2,930	605	650	138	141
4			1,050	450	275	190	161[b]
5			960	450	295	165	144
6	Two	Biplane	3,800	1,270	1,510	145	154
7	Three	Monoplane	2,600	1,290	835	144[c]	125
8			2,500	1,290	854	146	127
9			1,800	675	490	135	121
10		Biplane	3,700	1,680	1,250	136	124

[a] Post Office Department finds *average* speed of single-engine macines to be 104 m.p.h.
[b] Retractable landing-gear.
[c] Post Office Department finds *average* speed of three-engine airplanes to be 115 m.p.h.

horsepower. This does not seem to vary greatly with the size or power of the engine. By using this figure in an actual comparison later, we can determine whether increasing present speeds by using additional power is economical. If we acknowledge that the cruising speed required to meet competition, to operate with a profit and to sell air travel to the public must exceed 120 m.p.h., we can form some idea of the power required for future satisfactory flying equipment.

Let us assume that a landing speed of 65 m.p.h. is required. To meet this requirement without auxiliary airfoils, a wing loading of not more than 15 lb. per sq. ft. is necessary. The aerodynamic-efficiency factors of 10 commercial air transports now being operated were calculated and are given in Table 2.

Because of variations in performance and weights obtained on the various models and on different airlines, accurate generalizing is impossible. However, use of the following factors as average figures for future models seems reasonable:

(1) Single-engine monoplanes, without retractable landing-gear	145
(2) Two-engine monoplanes or biplanes, large size only, without retractable landing-gear	155
(3) Three-engine monoplanes or biplanes, large size only, without retractable landing-gear	135

With speeds to be expected in the near future, a completely retractable landing-gear can be estimated to increase the aerodynamic-efficiency factor by approximately 10 per cent.

The ratio of payload to gross weight also is necessary. This depends upon structural efficiency and equipment and must be determined on the basis of a dozen other variables, such as speed, price, passenger accommodations, engine and arrangement. The ratio is affected very little by the number of engines, probably because the proportion of horsepower required does not depend upon size so much as upon performance.

Let us assume a payload of 22 per cent for our power calculation. If we now assume a fictitious airplane to have a payload of 2500 lb. and an average speed of 125 m.p.h., then we have

Gross weight (2500/0.22) = 11,400 lb.	(12)
Wing area (11,400/15) = 760 sq. ft.	(13)
Cruising speed = 140 m.p.h. (from chart)	(14)
Maximum speed (140/0.85) = 165 m.p.h.	(15)

Actual Horsepower Required

For single-engine monoplane $760 \times (165^3/145^3) = 1110$ hp.	(16)
For two-engine monoplane or biplane $760 \times (165^3/155^3) = 910$ hp.	(17)
For three-engine monoplane or biplane $760 \times (165^3/135^3) = 1380$ hp.	(18)

Most Desirable Number of Engines

The example chosen is intermediate between two groups of airplanes now being used. The calculations given explain why the two such distinct groups exist. The largest power available from standard acceptable engines is about 700 hp. This power is not enough in one unit to give single-engine planes of the size quoted an acceptable performance; therefore, the smaller group of airplanes must be restricted to smaller payload, less speed or both.

On the other hand, if two or more engines are installed, economical operation makes essential either carrying more payload or higher speed or both. Thus the machines in the larger group must necessarily fall into the multi-engine class. The question of which class of airplane to use on a given airline depends entirely upon operating conditions, the most important factor being traffic density. To operate one 40-passenger airplane once a month would be as economically foolish as to operate a 4-passenger machine every 10 min.

We have had examples of airplanes in the upper class powered with one engine, but as passenger carriers they have proved to be distinct failures as regards both economy and speed. Machines have also been built in the lower class with two or more engines for supersafety, but they also have failed from the economy, speed and weight viewpoints.

As the speed competition and traffic density increase, the proportion of airplanes for which more than one engine is economically imperative can be expected to increase. This brings up the question, Shall the airplane be equipped with two, three or more engines? Arguments can be presented from many viewpoints, and, naturally, different conclusions will be reached.

Safety.—Some years ago the relative chances of forced landings by airplanes equipped with one, two, three and four engines was calculated carefully by T. P. Wright, of the Curtiss Aeroplane & Motor Co. The reasonable assumption was made that each multi-engine machine had sufficient power to maintain a specific safe altitude with one, but not more than one, engine dead. The calculation was made on a theoretical basis and showed that the relative chances for forced landing with engines of poor reliability were as follows:

One engine ÷ two engines = 300 ÷ 9 = 33.3:1
Three engines ÷ two engines = 26 ÷ 9 = 2.9:1

With more reliable engines the calculated ratios in favor of two engines are much greater.

The theory has now been amply supported in practice by the relative safety records of the various types. One operator using two-engine airplanes exclusively has avoided at least five forced landings due to engine failure, most of which would have ended in serious damage. Had these machines been equipped with only one engine, the probable number of forced landings would have been two or three. Such records cannot fail to have an influence on the insurance rates or crash reserve, and therefore upon costs and earnings.

Aerodynamic Efficiency of Multi-Engine Planes

Performance.—Speed and climb, other things being equal, depend upon aerodynamic-efficiency. A glance at the list of the aerodynamic-efficiency factors will show that, with the exception of one single-engine airplane with retractable landing-gear, the twin-engine biplane is the most efficient now being operated. This is brought out in another manner when the figures for payload per horsepower are listed. These are as follows for the two and three-engine transports now in operation:

	Payload per Horsepower, Lb.
Two-Engine Biplane	2.99
Three-Engine Monoplane	2.02
Three-Engine Monoplane	1.94
Three-Engine Monoplane	2.67
Three-Engine Biplane	2.20

When the cost of engine maintenance, fuel, oil and other charges depending upon the horsepower are considered, the difference in earnings is enormous.

The power required for flight on one engine in a two-engine airplane is somewhat more than 100 per cent in excess of the minimum required for maintaining a specified minimum altitude. That required for flight with two engines in a three-engine machine is only somewhat more than 50 per cent in excess of that required to maintain altitude. However, as the demand for performance has risen, we find that, even to meet present requirements, the necessary power is automatically 100 per cent in excess of that required to maintain

some altitude. The horsepower required, then, is determined largely by performance rather than by single-engine ceiling.

Comfort.—The factors that affect passenger comfort are primarily noise, vibration, space, seats, ventilation, temperature and humidity. If the airlines are to attract the repeat patronage upon which their whole revenue depends, they must give the greatest possible consideration to every one of these factors. Noise and vibration depend upon the engine arrangement and the care taken in reducing them.

In measurements of sound reduction made by the Bureau of Standards, the average cabin-sound reduction measured on two three-engine airplanes was 15 decibels, and the average for two two-engine planes was 30 decibels. The difference is largely explained by sound transmitted through the structure from the center engine. Vibration probably depends even more upon the presence of a center engine. The tests further showed that, at the noise and vibration level found in certain machines, an actually nauseating effect was produced upon some of the passengers. Certainly no greater hindrance to air travel exists than the fear of air-sickness that is engendered by passenger experience.

Weight.—An important consideration from the designer's viewpoint is the structural efficiency of mounting heavy units, such as engines, directly on the structure that is their ultimate support rather than through an intermediate fuselage structure. The value to the operator in increasing payload is indirect but is extremely important.

When superior visibility and comfort for the pilot and simplification of engine controls and accessories are added to the safety, performance, comfort and weight arguments, little doubt remains of the superiority of the two-engine airplane for transport service when the passenger traffic is such as to justify payloads in excess of approximately 2200 lb.

Relation of Wing Area and Speed

Wing Area.—The wing area of an airplane usually is determined upon the basis of weight, by considerations of landing speed and, in many cases, also of gliding or climbing angle with the engine or one engine dead. These are in the realm of indirect and unmeasurable variables, but they can exert considerable influence on earnings. This is in addition to the fact that the Department of Commerce probably will, in the future, enforce its present maximum landing-speed requirement for transport airplanes more rigorously than it does at present. Further, it seems only a question of time and experience before the insurance companies will set their rates, or the operators their reserves, on the basis of a "figure of merit" determined by safety records. Certainly landing speed and the number of engines will be two of the important determining factors. When the sums paid to the insurance companies are considered, these factors must be of economic importance. It is even possible that the public will become educated to the importance of landing speed and number of engines, as many railroad passengers have been educated to the safety value of steel coaches.

While a large wing area decreases landing speed, it does not necessarily decrease cruising speed. The effect is not nearly so great as would be indicated by the speed equation quoted. The theory explains it, that more wing area not only decreases the angle of flight but otherwise improves the aerodynamic efficiency. The theory is borne out by the fact that some airplanes have actually flown faster when wing area was added.

The subject of wing area brings up the moot question of wing arrangement. A popular view that monoplanes are faster than biplanes is current. Other things being equal, this is not true. In racing airplanes, where speed is the all-important consideration, a decisive trend in favor of either arrangement has not been evident. In every case where monoplanes are used exclusively for racing, the wings are wire braced. Alternative possibilities for wing support are strut bracing or cantilever construction; and neither of these, even though the appearance is neater, is acceptable from the speed viewpoint. The cantilever arrangement in particular seems contrary to the trend in high-speed military and racing machines in having the largest, rather than a smaller, section of the wing within the slipstream. However, the real decision between monoplanes and biplanes rests with wing loading and size. If the airplane is to be large and the wing loading is to be such as to give reasonable altitude performance and landing speed, it must either be a biplane or have a span and length such as to make hangar provision economically impractical. If the airplane is to be in the smaller class, a monoplane probably is indicated.

In addition, the wing-arrangement problem is tied up with the number of engines that should, or are to, be used. To have as little frontal area as possible in the slipstream is very desirable, aerodynamically, and in the smaller single-engine type this can best be accomplished with the externally braced structure. In the larger class having outboard engines approximately in line with a wing for minimum interference drag has been found also of utmost importance. This, in conjunction with the desirability of keeping engines accessible from the ground, leads to a decision in favor of either a wire-braced monoplane or biplane when efficient performance is the determining factor.

How Size and Shape Affect Earnings

Size and Shape.—In addition to wing area, the aerodynamic efficiency is controlled by size and shape of the airplane. The size inside depends upon the number of passengers and the space and facilities provided for their comfort. The design engineer's first tendency is to decrease the inside dimensions in favor of speed. However, the ultimate competition factors among various airlines and between airlines and other transportation services would inevitably seem to be comfort and service in addition to speed and fare. This will be even more marked if the fare competition is eliminated by public control, such as the Interstate Commerce Commission is already seeking and is likely to get over interstate motorcoach lines. Therefore, space accommodations are not likely to diminish.

Outside size is determined by the inside size and the type of construction. More actual cubic content inside a fuselage of given outside dimensions can be obtained when the construction is monocoque than in other types. However, because of the circular or elliptical shape, whether more passengers can be comfortably seated is very questionable.

Shape of the exterior is of extreme importance in determining speed and, together with engine arrangement, offers the most promising field for improvement. Many of the new military machines and some commercial airplanes now have a metal monocoque construction to achieve efficient aerodynamic shape, but no fundamental reason makes impossible the obtaining of shapes that are just as efficient by other means. This is brought out, if not proved, by the fact that the large majority of racing machines are of other construction. Where economy of manufacture and maintenance is a consideration, little doubt can exist in making a choice between the two types until the quantity of production has been multiplied many times.

Interest Charges on Flying-Equipment Investment.—The charge for money used in purchasing flying equip-

ment does not properly belong with operating expense, but nevertheless it is dependent upon the price paid for such equipment and therefore should be included in a comparative estimate of earnings at the beginning of operations. If this money is obtained through loans or bonds, it can be expected to cost about 6 per cent, and for the first year of operation it will come to nearly that amount. If the money is obtained through an issue of capital stock, the accounting of expenses does not include a corresponding charge and just how the charge should be handled is a matter of much controversy. However, if transport operation is to pay dividends, the decision on purchases should take into consideration an initial interest charge on investment of at least 6 per cent. Balanced against this is the fact that large reserves for depreciation, maintenance and insurance are set up during operations, and, if these are at all conservative, the going condition of the airline should require very little in the way of borrowed funds. Let us, then, neglect this item.

Airline-Operation Account with Average Airplane

Let us now consider a specific example of an airline several years hence in which all the conditions are as nearly average as possible. If we keep all conditions the same except one and those depending upon that one and try comparative values of that figure, we get a clear picture of the effect of the changed condition upon earnings.

Airplane Conditions	
Cruising Speed, m.p.h.	140
Payload, lb.	2,500
Passenger Seats	12
Initial Cost of Airplane, with Complete Equipment less Engines	$25,000
Economic Life of Airplane, hr.	5,000
Engine Horsepower, Actual	900
Engine Cost, (900 × $10.60)	$9,540
Engine Life, hr.	2,500
Crew	1
Operating Conditions of Fictitious Average Airline	
Potential Passengers per Year, Both Ways,	20,000
Route Miles, One Way	1,200
Average Miles per Stop	200
Number of Stops	7
Average Time Lost per Stop, min.	10
Average Use of Capacity, per cent	60
Ratio of Runs Completed to Runs Scheduled	0.95
Calculations, All Approximate	
Schedule Seats Required per Year, Both Ways, (20,000/0.60)	33,000
Number of Schedules per Year, Both Ways, (33,300/12)	2,780
Airplane-Miles per Year, Total Scheduled, (1,200 × 2,780)	3,330,000
Average Schedules per Day, Both Ways, (2,780/365)	7.61
Maximum Schedules per Day, Both Ways	8
Average Miles per Day per Airplane, when Operating, (1,200/2)	600
Number of Airplanes Required (This Also Allows for Overhaul) (8 × [1,200/600]) = 16; say, for spare,	17
Average Number of Airplanes in Service (7.61 × [1,200/600])	15.22
Schedule Runs per Airplane per Year ([2,780 × 1,200] / [17 × 600])	328
Total Airplane-Miles per Year (600 × 17 × 328 × 0.95)	3,180,000
Average Speed (from chart), m.p.h.	125
Total Hours per Year (3,180,000/125)	25,400
Average Hours per Year per Airplane (25,400/17)	1,500
Average Number of Passengers per Airplane (12 × 0.60 × [15.22/17])	6.45
Revenue from Passenger Operations (3,180,000 × 0.050 × 6.45)	$1,025,000
(*a*) Depreciation on Airplane and Equipment (0.0002 × 25,000)	$5.00
(*b*) Depreciation on Engine(s) (0.0004 × 9,540)	3.82
(*c*) Gasoline and Oil (0.011 × 900)	9.90
(*d*) Spare Parts, Material and Labor for Maintenance, Repair and Overhaul of Airplane (0.000095 × 25,000)	2.38
(*e*) Spart Parts, Material and Labor for Maintenance, Repair and Overhaul of Engine(s) (0.0055 × 900)	4.95
(*f*) Insurance or Reserve for Fire, Crash, Theft, Windstorm and the Like (0.000058 × 34,540)	2.00
(*g*) Insurance for Public Liability, Passenger Liability, Property Damage and the Like (0.000065 × 34,540)	2.24
(*h*) Pay and Compensation Insurance for Crew	7.50
Net Operating Expense per Hour (One Airplane)	$37.79
Total Operating Expense (37.79 × 25,400)	$959,900
Gross Earnings from Passenger Operations	$65,100

Having set up the account for operation with an "average prospective airplane," let us see the effect of changing some of the conditions to learn how much the earnings are affected thereby.

How Changing Cruising Speed Affects Earnings

Cruising Speed, by Changing Horsepower.—If all other characteristics of the airplane are kept the same, all operating conditions are the same and the cruising speed is changed by altering the horsepower, we have the following:

Cruising Speed, m.p.h.	140	120	160
Average Speed, m.p.h.	125	109	141
Actual Power Required, hp.	900	575	1,350
Cost of Engine(s)	$9,540	$6,100	$14,300
Passenger Fare per Passenger Mile	$0.050	$0.0475	$0.0525
Cost of Airplane and Engine(s)	$34,540	$31,100	$39,300
Time Operated per Year, hr.	25,400	29,200	22,500
Revenue[a]	$1,025,500	$974,200	$1,076,800
(*a*)	$5.00	$5.00	$5.00
(*b*)[a]	3.82	2.44	5.74
(*c*)[a]	9.90	6.32	14,85
(*d*)	2.38	2.38	2.38
(*e*)[a]	4.95	3.16	7.42
(*f*)[a]	2.00	1.80	2.28
(*g*)[a]	2.24	2.02	2.56
(*h*)	7.50	7.50	7.50
Net Operating Expense per Hour	$37.79	$30.62	$47.71
Net Operating Expense per Year[a]	$959,900	$894,100	$1,073,500
Gross Earnings from Passenger Operations	$65,100	$80,100	$3,300

[a] Items changed.

Conclusion 1.—In the case of the fictitious airplane and airline chosen, and in the absence of speed competition, using a low speed is better economy, if any increase must be obtained by use of more power alone.

This depends, of course, upon the fare that can be charged and also upon the existing or the anticipated competition, both of which can be determined by the individual operator more accurately than they can be here. In a precise calculation, such factors as change in the cost of the airplanes, effect of the change in the number of schedules per day and the like would have to be included.

Cruising Speed, by Changing Aerodynamic Efficiency.—If the aerodynamic efficiency alone is changed without altering the price of the airplane or any of the other variables, we have the following:

Cruising Speed, m.p.h.	140	120	160
Average Speed, m.p.h.	125	109	141
Time Operated per Year, hr,	25,400	29,200	22,500
Revenue[a]	$1,025,000	$974,200	$1,076,800
(*a*)	$5.00	$5.00	$5.00
(*b*)	3.82	3.82	3.82
(*c*)	9.90	9.90	9.90

(*d*)	2.38	2.38	2.38
(*e*)	4.95	4.95	4.95
(*f*)	2.00	2.00	2.00
(*g*)	2.24	2.24	2.24
(*h*)	7.50	7.50	7.50
Net Operating Expense per Hour	$37.79	$37.79	$37.79
Net Operating Expense per Year[a]	$959,900	$1,103,500	$850,300
Gross Earnings from Passenger Operations	$65,100	$129,300[b]	$226,500

[a] Item changed.
[b] Deficit.

Conclusion 2.—It is excellent economy to increase speeds by improving the aerodynamic efficiency of the flying equipment. On the basis of the assumptions made for the average airplane, the earning value per year of 1 m.p.h. increase in cruising speed approximates $500 per airplane.

Actually, to improve aerodynamic efficiency, increasing the weight, manufacturing cost and maintenance cost is necessary. These, however, would be largely balanced by the reduction in maximum fuel requirements for a given run. In a more precise estimate these factors should be considered.

When Replacement of Equipment Pays

Cruising Speed and Passenger Seats, by Replacement of Equipment.—We can now determine whether replacing equipment that is now in operation with a new type that is more economical to operate but may or may not increase earnings is economically sound and advisable. Let us assume that the airplanes now being operated conform with the following conditions:

Average Speed, m.p.h.	109
Passenger Seats	10
Pilots	1
Engines	3
Wing Area, sq. ft.	600

Then:

Payload, lb.	2,200
Gross Load, lb. (2,200/0.22)	10,000
Cruising Speed, m.p.h.	125
Approximate Maximum Speed, m.p.h. (125/0.85)	147
Actual Horsepower Required ($600 \times [147^3/135^3]$)	775
Cost of Engines ($775 \times \$10.60$)	$8,200
Cost of Complete Airplane ($20,000 + $8,200)	$28,200
Number of Airplane Hours per Year ($25,400 \times [12/10] \times [125/109]$) =	$34,900
Number of Airplanes Required ($17 \times [12/10] \times [125/109]$) 23.4, say	23

Let us assume that the airplanes being operated have had an average time of 2400 hr. and that the engines have had 2500 hr. This assumption is made because probably the most advantageous time to replace equipment is at the end of the economic life of the engines. To form a just picture of the advantages and disadvantages of scrapping old equipment for replacement with new, the comparative costs over several years must be shown.

If the original cost of the old airplane was $20,000 and the depreciation was based on 5000 hr., the present value per airplane under the conditions assumed is $20,000 \times (2600/5000) = \$10,400$.

If the net scrap value is estimated at $2,000, then the cost of scrapping will be $10,400 - $2,000 = $8,400 per airplane, or $23 \times \$8,400 = \$193,000$.

Let us amortize this cost during the first year of operation with new equipment so that the account will stand about as shown in Table 3.

TABLE 3—COMPARISON OF OPERATING COSTS WITH OLD AND NEW EQUIPMENT

	With Old Equipment Each Year	With Replacement First Year	With Replacement Succeeding Years
Revenue	$1,025,000	$1,025,000	$1,025,000
(*a*)	$4.00	$5.00	$5.00
(*b*)	3.28	3.82	3.82
(*c*)	8.53	9.90	9.90
(*d*)	1.90	2.38	2.38
(*e*)	4.26	4.95	4.95
(*f*)	1.64	2.00	2.00
(*g*)	1.83	2.24	2.24
(*h*)	7.50	7.50	7.50
Net Operating Expense per Hour	$32.94	$37.79	$37.79
Net Operating Expense per Year	$1,149,600	$959,900	$959,900
Amortization of Old Equipment		193,000	
		$1,152,900	
Gross Earnings from Passenger Operations	$124,600[a]	$127,900[a]	$65,100

[a] Deficit.

Thus the operating loss for the first year will be only slightly more than the loss each year with the old equipment, while good operating profits are shown during all succeeding years.

Another factor entering into a decision on scrapping present airplanes is the additional cash outlay required. Funds for replacing equipment or buying new equipment at any time must often be obtained from or through the banks. This is not easily done unless the prospects for future earnings are good. If these earnings are impossible with present equipment, the sooner it is replaced and the business put on a profitable basis the better.

Conclusion 3.—Replacing obsolete flying equipment with a more efficient type before the obsolete equipment has been used for its full life is economically sound and advisable, provided the total operating costs of the new equipment are, as in the foregoing example, enough less to justify such a change.

Value of Increasing Passenger Seats

Number of Passenger Seats, by Changing Design Efficiency, Materials or Equipment without Price Change.—The number of passenger seats usually is controlled by the weight and space provisions in the initial design. If we assume that, without altering the other factors, we can control the number of seats, we have the following:

Number of Seats	12	10	14
Time Operated per Year, hr.	25,400	30,500	21,800
Revenue	$1,025,000	$1,025,000	$1,025,000
(*a*)	$5.00	$5.00	$5.00
(*b*)	3.82	3.82	3.82
(*c*)	9.90	9.90	9.90
(*d*)	2.38	2.38	2.38
(*e*)	4.95	4.95	4.95
(*f*)	2.00	2.00	2.00
(*g*)	2.24	2.24	2.24
(*h*)	7.50	7.50	7.50
Net Operating Expense per Hour	$37.79	$37.79	$37.79
Net Operating Expense per Year[a]	959,900	1,152,600	823,800
Gross Earnings from Passenger Operations	$65,100	$127,600[b]	$201,200

[a] Item changed.
[b] Deficit.

Actually, the assumption that the number of seats can be changed as much as this without changing the price of the airplane and its performance is not valid. If such a change were possible, the value per year to the operator of each additional passenger per airplane would be approximately (177,700/151) × (100/17) × (14 − 103 = $4,840; or, at 200 lb. for a passenger and his baggage, this would be $24.20 per lb. This shows clearly how expensive adding extra equipment is if it involves any reduction of passenger capacity. For the manufacturer of transport equipment to spend $24.20 × (5000/1500) = $80.70 per airplane for each pound of weight saved also would seem worth while if this weight could be saved by design efficiency or the use of special materials.

Let us investigate further to see if it would be worth while for the operator to spend this amount as additional price for the airplane and engine.

Number of Passenger Seats, by Changing Design Efficiency or Materials, with Price Change.—On the basis of an additional price paid for airplanes, as calculated in the foregoing paragraph, the difference in value made by two passengers would be $80.70 × 2 × 200 = $23,230 per airplane. This is obviously too high, since it mounts to very nearly the original price paid for the 12-passenger machine with engine. Let us then suppose that the manufacturer, to sell his equipment at all, designs and sells it on the basis of one-third this value per pound saved, or about $80.70/3 = $26.90. If the cost of weight change is divided proportionately between airplane and engine, the account will stand about as shown in Table 4.

Thus, to buy airplanes with larger capacities is more profitable, provided the amount paid per pound additional does not exceed about $25. From these figures the fact is obvious that, if the manufacturer's price were much greater than $25 per pound of additional payload, the operator would lose money. Other factors, such as schedule frequency and corrections for independent charges which depend on the number of airplanes, should be taken into account. These must be fixed by the original operator when making comparative estimates.

On the basis of the foregoing caluculations, the manufacturer would charge about $11,000 for the weight saved to make provision for carrying two additional passengers. This rate is just about twice the charge he makes per passenger for the original seating capacity. This is reasonable, because, as structure is made lighter, the saving of further weight becomes increasingly difficult and before very long absolutely impossible.

TABLE 4—EFFECT OF INCREASING NUMBER OF SEATS ON OPERATING EXPENSES, IF COST OF ADDITIONAL WEIGHT IS DIVIDED PROPORTIONATELY BETWEEN THE AIRPLANE AND THE ENGINE

Number of Seats	12	10	14
Time Operated per Year, hr.	25,400	30,500	21,800
Cost Difference	0	$10,760[a]	$10,760[b]
Cost of Airplane and Engine	$34,540	$23,780	$45,300
Cost of Airplane (72.4 per cent)	$25,000	$17,230	$32,800
Cost of Engine(s) (27.6 per cent)	$9,540	$6,550	$12,500
Revenue	$1,025,000	$1,025,000	$1,025,000
(*a*)[c]	$5.00	$3.45	$6.56
(*b*)[c]	3.82	2.62	5.00
(*c*)	9.90	9.90	9.90
(*d*)[c]	2.38	1.64	3.12
(*e*)	4.95	4.95	4.95
(*f*)[c]	2.00	1.37	2.63
(*g*)[c]	2.24	1.54	2.94
(*h*)	7.50	7.50	7.50
Net Operating Expense per Hour[c]	$37.79	$32.98	$42.60
Net Operating Expense per Year[c]	$959,900	$1,005,900	$930,700
Gross Earnings from Passenger Operations	$65,100	$19,100	$94,300

[a] Decrease.
[b] Increase.
[c] Items changed.

Conclusion 4.—Seating as many passengers as possible in each airplane is economically advantageous as long as a satisfactory schedule frequency can be maintained and provided the price paid per pound of additional payload does not exceed a fixed amount that can be determined by an analysis such as that presented. In the example chosen as an average airplane on an average airline, this price approximates $25 per lb.

General Conclusion

By following a similar method of attack, determining also the effect on earnings of such variables as number of engines, number of crew members, potential passengers per year, average time lost per stop, average percentage of use of capacity and number of hours per year per airplane becomes a simple matter.

No claim has been, or can be, made as to the accuracy of the factors or the figures used in these estimates, but they have been kept as close as possible to the average for prospective operation. We hope that we have helped to remove the selection of airplanes from the realm of opinion by presenting a method of attack by which an individual operator can compare alternative types of equipment and, in conjunction with a knowledge of his specific airline conditions, decide upon a type that will produce the maximum earnings.

Legislative Regulation of Motor-Vehicles

Discussion of Thomas H. MacDonald and J. T. Thompson's Transportation Meeting Paper[1]

CHAIRMAN M. C. HORINE[2]:—Governmental authority over the use of public roads is necessary for the protection of the lives and property of the citizens, for safeguarding the public investment in roads and bridges, for financing the construction and maintenance of highways and for assurance to all of the people equal rights to the fullest and fairest utilization of those primary and greatest of all public works—the public highways. This control takes three main forms: restriction, taxation and regulation.

Restriction properly limits the size, weight and tire loading of the vehicles as related to structural strength and widths of roads and bridges, curvatures of the roads, overhead clearances and conditions of traffic.

Special taxation seeks to raise the necessary funds for highway work by assessing individuals or corporations according to the extent of the use or benefit they derive.

Regulation is necessary to prevent abuse of the facilities by reckless or greedy users and is concerned with such matters as speed, lighting and safety equipment, qualifications of drivers, financial responsibility of owners, safe and convenient handling of traffic and the prevention of unfair practices with respect to the patrons of highway transport or as toward one another by those who use the public highways for the conduct of their business.

The motor industry has no quarrel with these proper governmental functions, whether exercised by Federal, State or Provincial or municipal authority; but when legislatures, commissions and the police pervert their authority to achieve other ends or where they exercise it without using diligence to learn or without the sincere purpose to apply the facts, then we feel that we have a right to complain. Were the facts not available, some excuse might exist for such misdirection of authority; but such is not the case. Through the untiring zeal of its public-spirited personnel and the true scientific spirit of honesty with which it has prosecuted its studies, tests and observations, the Bureau of Public Roads has developed certain basic facts relating to road structures, impact effects of vehicular traffic, economic relationships between road costs and taxation and the influence of traffic on road design. The results of these studies constitute the most complete and authoritative thesis on the subject in existence and are of public record and public property.

The paper prepared by Mr. MacDonald, under whose direction this monumental work has been accomplished, in collaboration with Professor Thompson, shows the limitations within which they believe motor-vehicle manufacturers must confine their vehicles so as to meet the requirements of the most economic utilization of modern highways. The recommendations are for regulations under which we might equitably conduct motor transport. But how little attention has been paid to the facts of the situation by our lawmakers and administrators.

Suggested Exception to Length Limit

T. C. SMITH[3]:—Mr. MacDonald has presented a very constructive, conservative and reasonable group of suggestions covering motor-vehicle widths, heights, lengths, speeds and loads. These items cover a broad field, and the recommendations presented undoubtedly will result in greater consideration by legislative bodies toward much needed unification of State requirements covering them.

I hope that Mr. MacDonald will favor us at as early a date as may be convenient with more information of the same type covering such additional items as motor-vehicle lighting requirements, permissible overhang of loads, brake requirements, lengths of drawbars and trailer-connection requirements.

One detail with respect to length of single vehicles and combinations of vehicles seems worthy of consideration. Public-utility companies must deliver poles from the railroad stations to the pole locations along the highways. These poles vary in length from approximately 20 to about 65 ft. Since the poles cannot be telescoped or dismantled, as may be possible with other kinds of loads, they must be transported over the highway using a combined length of vehicle and trailer that in some instances exceeds the specified total of 35 ft. for a single vehicle or 65 ft. for a combination of vehicles. The same situation is encountered by contractors hauling piling, girders and the like. With this in mind, I suggest the addition, to each of the first two items under the length recommendation, of a statement to the effect that

> Exceptions are made for vehicles hauling poles, piling, pipe, girders and similar loads which cannot be dismantled for transportation.

Danger of Too Much Regulation

J. F. WINCHESTER[4]:—The Bureau of Public Roads, through this paper, again demonstrates its willingness to advance and promulgate ideas that should benefit the Nation as a whole. We, as citizens and members of a responsible society, should take up the cudgels and hew away through demand for action on uniform motor-vehicle laws.

The Society has a committee studying many of the points discussed in this paper. It is cooperating with certain other national organizations. As a member of that committee, I appreciate that to draw the Society into national or State politics would be a mistake, but I hope that the individual members of that committee will not rest content with merely drawing up a recommended practice but will continue to promote their proposals by taking an active interest in State organizations in their respective communities.

(*Concluded on p.* 492)

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[1] Published in the S.A.E. JOURNAL, November, 1932, p. 421. Mr. MacDonald is chief of the Bureau of Public Roads of the Department of Agriculture and Mr. Thompson is professor of civil engineering at Johns Hopkins University, Baltimore.

[2] M.S.A.E.—Sales promotion engineer, International Motor Co., Long Island City, N. Y.

[3] M.S.A.E.—Engineer, automotive equipment and construction apparatus, American Telephone & Telegraph Co., New York City.

[4] M.S.A.E.—Supervisor of motor-vehicles, Standard Oil Co. of New Jersey, Newark, N. J.

Six-Wheel Trucks

Discussion of Austin M. Wolf's Transportation Meeting Paper[1]

MERRILL C. HORINE[2]:—Considering the present paper and recollecting other papers on the same subject presented but a few years ago, one is impressed by the great progress that has been effected in a short time, thus proving that the open mind still rules in the truck industry. Nowhere is this better illustrated than in the abrupt reversal of sentiment concerning the interaxle differential or power divider.

Mr. Wolf has ably covered our present knowledge of torque phenomena on four-rear-wheel drive as well as the rapidly crystallizing opinion with respect to interaxle differentials. I would go farther and suggest that a very close relationship exists between these two subjects in that, after everything has been done to minimize load transference from one driving axle to another, due to torque reactions, the interaxle differential offers a further compensation to equalize axle loading. This gives torque balance as between the two axles and tends to prevent excessive torque reaction on the rearmost axle by preventing that axle from receiving an undue share of the torque from which the reaction arises, which seems to me to add one more reason for believing that, at least in heavy vehicles, the interaxle differential is here to stay.

Referring to the suspension of the Mack shaft-driven six-wheel bogies, Mr. Wolf at one point correctly mentions the use of "torque-rods," whereas elsewhere he inadvertently refers to them as "torque-arms"[3]. I believe that a clear distinction should be made between the torque-arm and the torque-rod. The former is a lever adapted to receive bending loads; a torque-rod, on the other hand, is a freely-pivoted strut adapted to receive loads in tension and compression. If torque-arms were used on the Mack rear bogie, an intolerable rearing effect would transpire from driving reaction because the moments at the points of attachment to the frame of the arms extending from the respective axles would be vertical and opposite. With torque-rods, however, these moments are horizontal under normal spring load and in the same direction and so exert no vertical reactions to cause rearing.

Mr. Wolf has done us a service, I think, in straightening out in Table 1 the classification of different six-wheel arrangements. I should like to suggest that, instead of designating the different groups by arbitrary Roman numerals, letter-combination symbols be used, selecting letters that would be suggestive of the construction designated. As we cannot use D for both Dead and Driving, I suggest D for Driving and I for Idling. Thus, we would have the arrangement shown in Table 1. The only trouble with this is that the last combination spells DID, while Mr. Wolf has carefully told us it is the only job which somebody didn't try.

TABLE 1—SUGGESTED SCHEME FOR CLASSIFYING THE DIFFERENT AXLE ARRANGEMENTS ON SIX-WHEEL TRUCKS

Group	Suggested Symbol	Front Axle	Center Axle	Rear Axle
I	IDI	Dead	Driving	Dead
II	IDD	Dead	Driving	Driving
III	DDI	Driving	Driving	Dead
IV	DDD	Driving	Driving	Driving
V	IID	Dead	Dead	Driving
VI	DID	Driving	Dead	Dead

JAMES W. COTTRELL[4]:—The question of interaxle differentials will find its answer in the experience of operators. Mr. Wolf has rendered a service in pointing out the difficulties of designing a suitable differential for interaxle service. To be truly effective it must be almost instantaneous in action and at the same time strong enough to transmit torque compounded through multi-speed transmissions. Such differentials act when a truck is grunting along in low low gear as well as when highballing along in overdrive.

B. B. BACHMAN[5]:—A wide difference of opinion still exists as to the desirability of, and the necessity for, a differential for use between the driving axles. Undoubtedly, the use of a normal differential construction would not be found satisfactory except under most limited conditions. The so-called "power divider," which name distinguishes a device that will permit differential action to accommodate speed differences of relatively small magnitudes, probably is a desirable feature if such a unit can be developed so as to provide a variable amount of power to the unit having traction.

A question has arisen in my mind as to whether we have definite information that the units that have been developed and proposed to perform this function really do it. I have felt that the operation of a unit of this sort was in part a function of speed and, therefore, that a construction which might operate satisfactorily when located in the conventional position on the rear-axle shaft operating at the speed of the wheels may not perform in the same way or satisfactorily when operating at the speed of the propeller-shaft.

Auxiliary Dead Axle Has Obvious Advantages

WILLIAM R. GORDON[6]:—The double-drive type of installation offers the major engineering problems at this time. However, from the marketing point of view we cannot afford to overlook the fact that possibly less than 1 per cent of the gross tonnage hauled on our highways today requires consideration of this type of equipment. The auxiliary dead axle of the trailing type has the obvious advantages of simplicity of construction, lower first cost and maintenance, very material reduction in unsprung weight and an absence of just those problems that contribute to the uncertainty of results in operating the double-drive type.

Our experience has been that gross loads exceeding 60,000 lb. can be handled efficiently by trucks in which the third axle has provision for changing the load-distribution ratio between the two rear axles; that is, throwing more of the weight from the trailing axle on the driving axle when unusual difficulties, such as snow and ice, are encountered. This, combined with

[1] Published in S.A.E. JOURNAL, October, 1932, p. 387. The author is a consulting engineer, New York City, and a member of the Society.

[2] M.S.A.E.—Sales promotion manager, International Motor Co., Long Island City, N. Y.

[3] See S.A.E. JOURNAL, October, 1932, p. 391, second column; also see correction in the S.A.E. JOURNAL, November, 1932, p. 437.

[4] M.S.A.E.—Technical editor, *Commercial Car Journal*, Philadelphia.

[5] M.S.A.E.—Vice-president, engineering division, Autocar Co., Ardmore, Pa.

[6] M.S.A.E.—Transportation engineer, Truck Equipment Co., Inc., Buffalo.

a high-traction type of differential in the truck-chassis driving-axle is, we believe, the most practical answer in by far the greater number of installations.

The third axle is in effect a close-coupled semi-trailer with the fifth wheel eliminated. The principle is as applicable to small as to large units, just as in trailer operation. Therefore, it deserves the careful consideration of engineers in combination with their lighter chassis models as well as for extremely heavy hauling. Legislation today is drastically restricting the gross loads that may be hauled on our highways and also the over-all length of a train of vehicles. The trailing type of auxiliary axle, because of its lighter weight as compared with trailer and double-drive rear axles, makes possible the carrying of a greater payload. Solution of engineering problems, such as the intermediate differential between double-drive rear axles, is important, but if eliminating such problems is possible in the large majority of installations by using a simpler design of auxiliary axle, it seems wise to do so.

As Mr. Wolf states, each type of equipment has its distinct field and the conventional two-axle truck will continue as before, but, when we consider that third-axle equipment offers advantages to the operator in a substantially increased gross-weight rating in 33 States and more to come, as against the general tendency to reduce gross-weight ratings on other equipment; the altogether handier and safer operation, as compared with trailers, and the reduced license and insurance charges, there seem to be good economic reasons why engineers should carefully consider and exploit its advantages.

T. C. SMITH[7]:—Two points with regard to the six-wheel type of equipment which I should like to discuss have to do with the use of one free rear axle and the providing of a drive for the front axle.

In experimenting with different types of free-axle attachment to the rear of conventional four-wheel trucks and also with free-axle equipment on Christie crawler-type outfits without the tracks, I have found that when the vehicle leaves the hard-surfaced road the danger of its being stuck always is present because of the action of the free axle in relieving part of the load from the driving axle. This free-axle support for a part of the load at the rear causes the driving wheels to lose traction and then to dig into soft ground, sand or mud so quickly that the vehicle is not usable under these conditions. The action is similar in soft snow.

If six-wheel vehicles are used on a private right-of-way, utilizing front-wheel drive is very desirable, as much for assistance in steering as for assistance in moving the vehicle. Experiments with six-wheel outfits without front-wheel drive showed that, where the ground was rather soft, the tendency was for the front wheels to push sideways while the vehicle went straight ahead and did not respond to the steering angle of the front wheels.

Undoubtedly a field exists for six-wheel vehicles, but seemingly it will be limited to the heavy-duty units.

Author's Comments on the Discussion

AUSTIN M. WOLF:—The universal-joint seems to be one of the units that is subjected to the greatest rigors. I explained in the paper about a 42-deg. interference angle with an interaxle differential[8], and I also wish to mention that sometimes longitudinal movement of as much as 5 in. is required of the splined joint. This is conceivable when one considers that the over-all lengths of the universal-joint shafts range from 26 to 36 in. This brings up the question whether the truck manufacturer should utilize the utmost flexibility of the bogie unit or if placing stops on the frame to limit the universal-joint angle to not more than approximately 30 deg. would be wiser.

The close relationship between torque reaction and the interaxle differential, mentioned by Mr. Horine, is an important consideration with a four-wheel-drive bogie unit. A study of Steeds' analysis[9] will bear this out. Some constructions permit a very wide variation in the load on the two axles when torque is impressed upon them. Should a construction be such that the forward axle is relieved of weight, the limit to torque with the interaxle differential that can be used will be the tendency of the front drive wheels to rise and slip. When no interaxle differential is used, the torque impressed upon the two axles will be proportional to the weights thereon. In the case of prevalent constructions, such as the Timken and the Mack, the axles are equally loaded, neglecting the results of flexure, and the static load is increased only by the transfer of weight from the front end of the vehicle.

I concur with Mr. Horine's suggestion to use letter symbols instead of the Roman numerals in Table 1, so that the designation will clearly indicate by itself the correct grouping. However, Group VI should be DII instead of Mr. Horine's DID. Unfortunately, this explodes his joke, but nevertheless the DID combination is possible and is another one that has not yet been tried. At a later date we might add it to Table 1 as Group VII.

Regarding Mr. Smith's comments on the use of four-wheel trucks with Christie crawler units, I believe that this type of equipment has outlived its usefulness and that the six-wheel truck will take its place. The low road-speed of a crawler attachment limits the vehicle in covering distances. For adverse conditions, the four rear wheels should be driven and should incorporate an interaxle differential. I have in mind a 7-ton crane unit that I am now mounting on a six-wheeler in place of the crawler equipment mentioned.

Mr. Smith is correct in the view that the front wheels should drive in very soft going so as to give the vehicle dirigibility when the front wheels are at an angle; otherwise they become merely points of support and their deflection does not react in steering the vehicle. This has occurred already with the four-wheelers.

Mr. Bachman mentioned the difficulties of building a suitable differential unit. Some present differentials would not function properly if they were to rotate at axle speed, but as they rotate at six to eight times axle speed, or whatever the reduction in the axle may be, they are operating at a low torque-loading and seemingly function satisfactorily.

I desire to reiterate that each type of six-wheeler should be applied to the field to which it is best suited; also that the four-wheeler is here to stay and that no one should attempt to force on a prospective purchaser a six-wheeler when the four-wheeler will do the work. Each individual problem should be analyzed carefully so that misapplication is impossible; otherwise confidence in our industry will be shaken.

[7] M.S.A.E.—Engineer, automotive equipment and construction apparatus, American Telephone & Telegraph Co., New York City.

[8] See S.A.E. JOURNAL, October, 1932, p. 397, under General Design and Requirements.

[9] See *The Automobile Engineer*, May, 1931, p. 172.

Automotive Research

Bending Moments in the Master Rod of a Radial Aircraft Engine

By C. F. Taylor[1]

HEREIN are presented the results of an investigation of the bending moments in the master rod of a radial aircraft engine by a graphical method, and a simple formula derived therefrom for approximating this moment in similar engines.

The bending stress in the master rod comes from turning moments about the crankpin axis caused by the action of the articulated rods due to gas-pressure and inertia forces and also by the inertia forces in the master rod itself. Charts are presented that show the magnitude and fluctuation of these turning moments.

Accurate computation of these moments involves much tedious work. A method of approximating them with sufficient accuracy for engineering purposes is given for the case of a nine-cylinder radial aircraft engine. The method is applicable also to non-radial engines and to radial engines having other than nine cylinders, but in these cases investigation of the turning moments due to the gas loads in certain cylinders seems advisable.

IN designing the master connecting-rod for a radial engine, one of the most difficult stresses to evaluate is the bending stress in the column of the rod. Accurate computation of this stress involves so much tedious work that it often is omitted from calculations and allowed for by approximations, or by arbitrarily increasing the column section-modulus at the inner end. In consequence, connecting-rod failures during the development stage have not been uncommon, and successful designs have been evolved largely by a process of trial and error.

The bending stress in the column of a master connecting-rod comes from a turning moment about the crankpin axis caused by the action of the articulated rods due to gas-pressure and inertia forces, and also from a bending due to inertia forces in the master rod itself. The character of these forces is shown in Fig. 1. The gas pressure on the piston of any one of the articulated-rod cylinders is transmitted through the articulated rod to the master rod, and this results in a couple acting around the crankpin axis, except when the axis of the hinge-pin lies on a straight line between the crankpin and the piston-pin axis. Obviously, the couple due to any one cylinder is the axial force in the articulated connecting-rod multiplied by its lever arm around the crankpin axis. The couple in the master rod due to the inertia force of the articulated rod and the piston is caused in a similar manner. In both cases the existence of such a couple is due to the fact that the line of action of the connecting-rod force usually does not lie on the line connecting the crankpin and the piston-pin axis. The bending due to the inertia of the master rod itself is common to all connecting-rods. For convenience, this latter has been designated the centrifugal moment, while the others are called respectively the gas-load moment and the inertia moment.

For convenience in handling, only those forces due to the reciprocating weights, that is, the pistons and upper ends of the articulated connecting-rods, should be included in the inertia-moment computation. The rotating weight of the articulated rods, that is, the weight of their lower ends and hinge-pins, is then included as part of the master-rod weight and is used in computing the centrifugal moment.

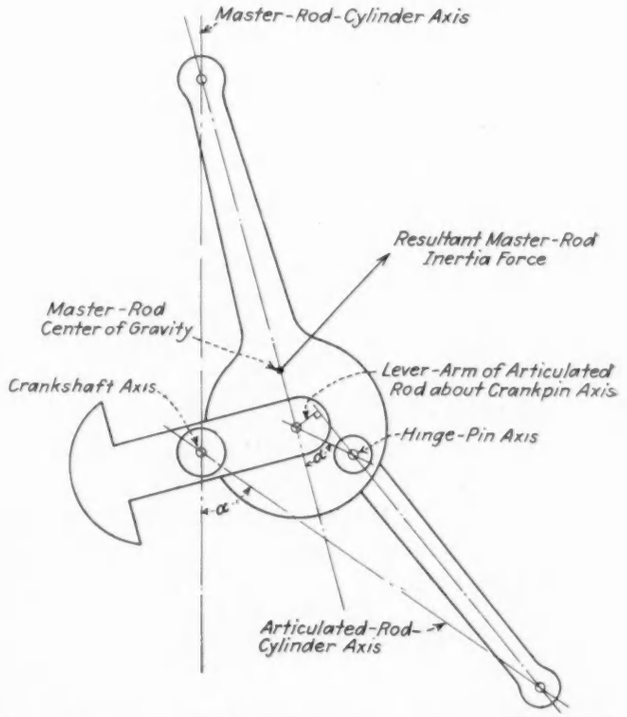

FIG. 1—DIAGRAM OF TURNING MOMENTS ABOUT THE CRANKPIN

[1] M.S.A.E.—Professor of aeronautic engineering, Massachusetts Institute of Technology, Cambridge, Mass.

To determine the magnitude of these moments on a typical radial-engine master rod, H. L. Van Alstyne, while a post-graduate student at the Massachusetts Institute of Technology, undertook their investigation in the case of the Wright J-5 nine-cylinder radial aircraft engine[2]. The principal statistics regarding this engine are given in Table 1, and a drawing of the master rod is shown in Fig. 2.

How Magnitude of Moments Was Determined

Graphical methods were used throughout. The positions of the crankpin, articulated pin and piston-pin centers were laid out with extreme care for positions of the crank at 10-deg. intervals. This was done on a sheet of aluminum with a sharp metal scribe, since the necessary degree of precision could not be obtained with pencil and paper. The layout so constructed gave the line of action and the lever arm of each articulated connecting-rod for each 10-deg. increment of crank rotation and also gave the positions of the center of gravity of the master rod and the piston-pins from which the accelerations could be computed.

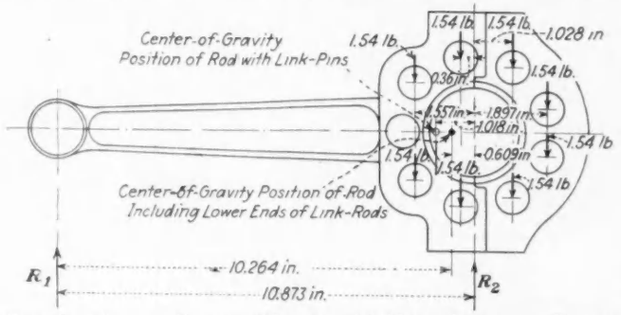

FIG. 2—MASTER ROD OF WRIGHT J-5 NINE-CYLINDER RADIAL ENGINE

This Drawing Shows the Redetermination of the Center of Gravity of the Master Rod When the Rotating Weight of Each Articulated Connecting-Rod Is Considered as Concentrated at Its Knuckle-Pin

The gas loads on the pistons were taken from a typical aircraft-engine indicator diagram, shown in Fig. 3. The pressure at each 10-deg. increment of crank rotation was computed for each piston. From this the force in the connecting-rod was computed and the couple caused by each rod was then found by multiplying the connecting-rod force by its lever arm around the crankpin center. All forces, lever arms and moments were carefully tabulated with the proper algebraic sign. The inertia forces acting at the center of each piston-pin were computed by assuming that the weight of the piston and the upper end of the connecting-rod, the so-called reciprocating weight, was concentrated along the axis of the piston-pin. The displacement-versus-time curve of the piston-pin was plotted for each cylinder and was graphically differentiated to determine acceleration. From the acceleration and weight the inertia force along the connecting-rod could be computed, and this, multiplied by the lever arm around the crankpin axis, was taken as the couple due to the inertia force in that particular cylinder. This quantity was also tabulated for 10-deg. increments for each of the articulated-rod cylinders.

The turning moments due to the inertia of the master rod itself were determined by plotting the path of the center of gravity of the master-rod assembly, including link-pins and articulated-rod rotating weights, and graphically determining the acceleration.

[2] See Thesis on An Investigation of the Bending Moments Imposed upon the Crankpin End of the Master Connecting-Rod of the Wright J-5 Radial Air-Cooled Aviation Engine, by H. L. Van Alstyne, 1929, on file in the library of the Massachusetts Institute of Technology.

TABLE 1—DATA REGARDING THE WRIGHT J-5 RADIAL AIRCRAFT ENGINE

Type	Radial, Air-Cooled
Number of Cylinders	9
Bore, in.	4.5
Stroke, in.	5.5
Piston Displacement, Total, cu. in.	788
Rated Brake Horsepower	210
Rated Speed, r.p.m.	1,800
Master Connecting-Rod Length, in.	10.873
Crank Radius, in.	2.75
Ratio of Crank Radius to Connecting-Rod Length	0.253
Average Radius of Link-Pins, in.	2.05
Weight of Piston Assembly, lb.	2.68
Reciprocating Weight of Piston Assembly and Upper End of Articulated Rod, lb.	3.44
Reciprocating Weight of Piston Assembly and Upper End of Master-Rod, lb.	3.90
Rotating Weight of Each Articulated Rod, lb.	1.54
Total Weight of Master-Rod Assembly, Including Pins but Not Articulated Rods, lb.	13.03

The results of this work are given by the curves in Figs. 4, 5 and 6. Fig. 4 shows the moments due to the gas forces in each of the eight articulated cylinders, together with the resultant turning moment from all of these cylinders combined. Fig. 5 shows similar curves derived from the inertia forces at 2000 r.p.m. of each of the eight articulated-rod cylinders and for their combination. Fig. 6 shows the centrifugal moment at 2000 r.p.m. and also the combined moments due to inertia forces, centrifugal force and gas forces respectively. The heavy line in this figure shows the resultant turning moment around the crankpin axis due to all three factors when the engine is running at 2000 r.p.m. with all cylinders firing. It was also considered of interest to determine moments due to cylinder pressures when the engine was not firing. These are shown in Fig. 7 for each of the eight articulated-rod cylinders and for all of them combined, and are designated as the moments due to pumping.

Discussion of Moments and Stress Computations

Of chief interest in Fig. 4 are the considerable magnitude of the moments due to the gas forces and the fact that the important maxima are largely due to the firing of individual cylinders. For instance, the maxi-

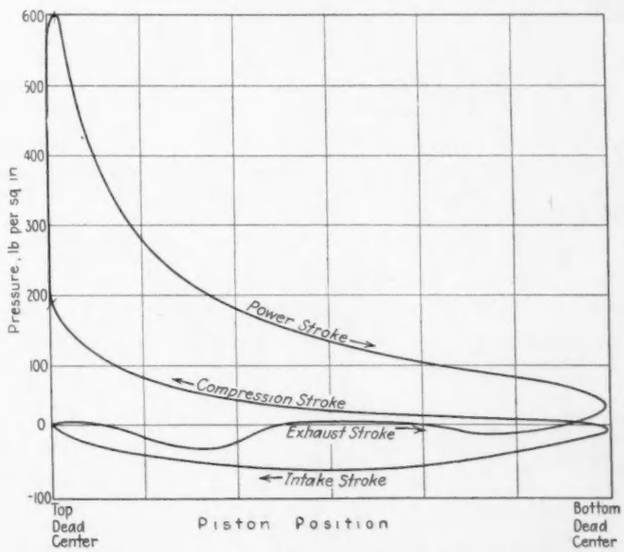

FIG. 3—TYPICAL AIRCRAFT-ENGINE INDICATOR-DIAGRAM FROM WHICH GAS LOADS ON THE PISTONS WERE TAKEN

mum at 490 deg. is due almost entirely to the action of cylinder No. 4, which starts its firing stroke at 480 deg. past the top-center position of the crank as referred to the master-rod cylinder, No. 1. The maximum moment comes substantially at the time of maximum pressure on the indicator card and amounts to 10,400 in-lb., while the combined moment from all cylinders at this particular point is 11,400 in-lb. No. 7 cylinder, which is symmetrical with No. 4 with respect to the axis of the master-rod cylinder, gives a nearly equal maximum moment, namely, 9900 in-lb., and the moment added by other cylinders at this point gives a maximum value for the combined gas-load moments of 12,500 in-lb. Probably the difference shown between the peaks at 250 and 490 deg. are within the limits of error of the methods used. Significant peaks in bending moment occur also at 100 deg. past top center, chiefly due to cylinder No. 3, and at 588 deg. past top center, chiefly due to No. 6.

In Fig. 5 the diagrams of the individual inertia moments at 2000 r.p.m. and also the resultant inertia moment repeat themselves from 360 to 720 deg. Fig. 6 shows the turning moment due to the inertia of the master rod itself at 2000 r.p.m. and also the resultants of the gas-load moments and inertia moments taken from Figs. 4 and 5. The heavy curve is the resultant of all three of these factors and represents the turning moment around the crankpin when the engine is running at 2000 r.p.m. with all cylinders firing.

From Fig. 7 it is very evident that bending moments from cylinders which are not firing are of minor importance compared with the moments due to firing, although their maximum is greater than the maximum moment due to centrifugal or inertia forces.

In computing stresses from these turning moments, the gas-load and inertia moments may be translated directly into equivalent loads at the piston-pin of No. 1 cylinder, at right angles to the master-rod axis. The connecting-rod column can be considered as a cantilever beam loaded at the end, and the stresses are easily computed. The stresses due to the centrifugal moment, however, are more difficult to compute, since they are a function of the detail design of the rod and its mass distribution. Fortunately, it can be shown that the centrifugal moment will always oppose the gas-load moment and therefore the worst condition will occur at full throttle—maximum gas load—when the centrifugal moment is at the minimum, as in a sudden acceleration from low speed. Therefore it is unnecessary to calculate the stress due to the centrifugal moment in the average case where the stress due to gas loads will be much greater.

Also noticeable in Fig. 6 is the fact that the inertia moment is insignificant as compared with the gas-load moment. In the case of aircraft engines and similar engines in which the reciprocating weights are relatively small and the maximum revolutions per minute are moderate, it may well be neglected. For engines having heavy reciprocating parts operat-

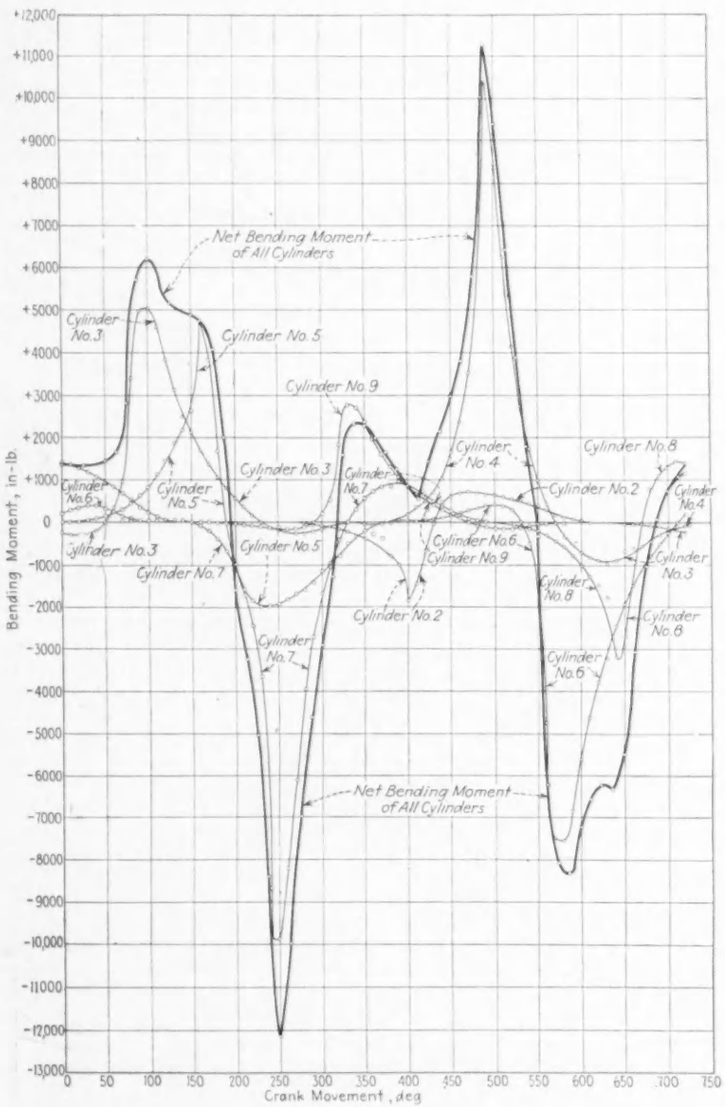

FIG. 4—TURNING MOMENTS EXERTED ON MASTER ROD BY GAS PRESSURE ON ARTICULATED RODS, ALL CYLINDERS FIRING

Engine Speed, 2000 R.P.M. The Master-Rod Cylinder Is Considered as Cylinder No. 1

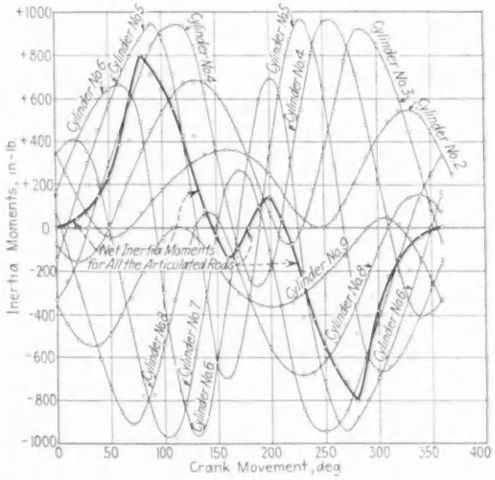

FIG. 5—TURNING MOMENTS EXERTED ON MASTER ROD CAUSED BY INERTIA FORCES OF ARTICULATED RODS

Heavy-Line Curve Indicates Resultant Turning Moment of All Eight Articulated Rods. Engine Speed, 2000 R.P.M.

ing at high speeds, the inertia moment might be important, but at present no practical example of this exists in radial-type engines.

Another significant fact deducible from Fig. 6 is that, since the maximum gas-load moment is due almost entirely to one cylinder, misfiring of this cylinder would decrease the value of this moment. On the other hand, misfiring of cylinders that were contributing moments in the opposite direction would slightly increase the resultant moment. In the case of the J-5 engine, however, all of the gas-load moments are in one direction at any of the significant maxima.

Computing the stresses in the J-5 master rod of Fig. 2, the maximum bending stress in the rod column due to gas loads alone is found to be 27,350 lb. per sq. in., and it occurs at the lower end of the column where it joins the big end. The maximum bending stress due to inertia is 1787 lb. per sq. in. The stress due to centrifugal action has not been determined but is evidently of a low order compared with the gas-load stress. If the mass distribution of the rod is neglected and the centrifugal moment is converted directly into an equivalent load at the piston-pin axis, the resulting fiber stress in the rod is 7360 lb. per sq. in. This probably is somewhat greater than the actual stress due to centrifugal action.

Since the maximum compressive stress in the smallest section of the rod column is about 16,000 lb. per sq. in., the greatest stress in the column evidently is due to bending.

Conclusions and Their Application

In drawing general conclusions from this investigation, it should be noted that in the J-5 engine, as in most radial engines, the angle α (Fig. 1) between the master-rod-column center line and a line connecting the crankpin center and any hinge-pin center is equal to the angle between the axis of the cylinder in question and the axis of the master-rod cylinder. This gives an appreciable lever arm about the crankpin axis when the crank is at the top-center position of the articulated-rod cylinder, which accounts for the relative importance of the gas-load turning moments. In V-type engines, where the hinge-pin is positioned on the master rod so that the lever arm is zero at the top-center crank position of the articulated-rod cylinder, the magnitude of the maximum gas-load moment is greatly reduced and the resulting bending stress may be less important than the bending due to inertia or centrifugal forces. Hence the following conclusions apply only to the case in which the cylinder angle and the hinge-pin-position angle are equal.

The foregoing has shown that the turning moment around the crankpin axis of a radial engine may involve a very appreciable stressing of the column of the master rod and that this stress should be allowed for in design. Maximum bending will occur when the gas loads are at the maximum and the inertia loads are at the minimum, and this occurs in a sudden acceleration from very low speed. Misfiring of the cylinders usually will not increase the maximum bending stresses.

Determination of the turning moments about the crankpin for all cylinders at all crank angles is a long and tedious process. Reduction of the work in this connection is desirable. This can be done by application of some of the general facts revealed in this report. If an engine is similar to the one investigated, that is, a nine-cylinder radial aircraft engine of reasonably conventional design, an estimation of the maximum bending moment around the crankpin that is probably within the necessary limits of accuracy for engineering purposes can be made by the following approximations:

(1) Since the maximum moment is largely the result of gas pressure on a single piston, this moment

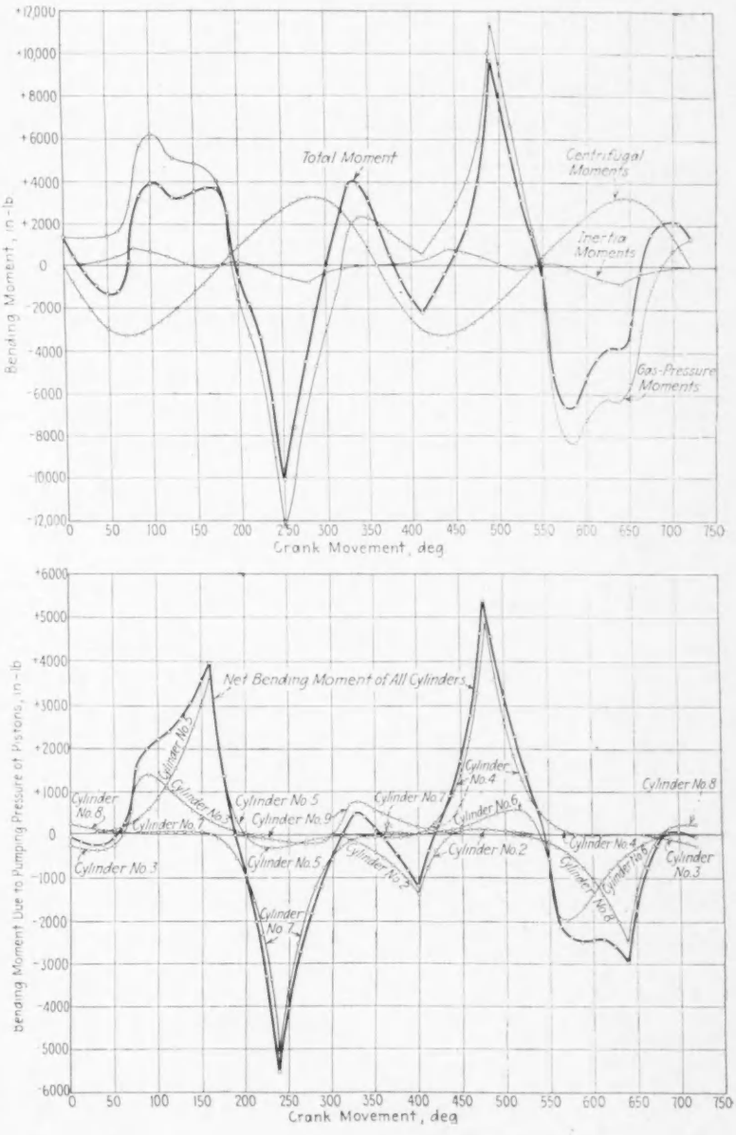

FIG. 6 (ABOVE)—CENTRIFUGAL TURNING-MOMENT AND COMBINED MOMENT CAUSED BY INERTIA FORCES, CENTRIFUGAL FORCE AND GAS FORCES

The Heavy-Line Curve Shows Resultant Turning-Moment around Crankpin Axis Due to All Three Factors. All Cylinders Firing

FIG. 7 (BELOW)—TURNING MOMENTS EXERTED ON MASTER ROD BY PUMPING PRESSURE ON PISTONS, ALL CYLINDERS MISFIRING

The Heavy Line Curve Indicates Resultant Turning Moment from All Cylinders. Engine Speed, 2000 R.P.M.

may be assumed to be proportional to the piston area or to the square of the cylinder bore.

(2) Within reasonable limits, the effective lever arm around the crankpin axis of an articulated rod near its top-center position is proportional to $Rr/L = r$, where
L = length of master rod, between centers
R = crank radius
r = average distance of articulated-pin axes from crankpin axis

The value of the foregoing expression for the J-5 engine is $(2.75 \times 2.05)/(10.87 - 2.05) = 0.064$.

Taking the J-5 engine of this report as a basis, the maximum turning-moment about the crankpin due to gas forces in a similar nine-cylinder radial engine may then be approximated by the following formula:

$$M_{max} = (12{,}250/0.523)\ (b^2/20.3)\ (Rr/L\text{-}r) = 944\ (b^2Rr/L\text{-}r) \quad (1)$$

where
b = bore of cylinder in inches
M_{max} = maximum turning moment in inch-pounds

For engines that are not similar to the one investigated as reported herein, that is, for non-radial types and for radials having other than nine cylinders, making an investigation of the bending moment due to gas loads apparently would be advisable. This can be greatly simplified by remembering the following facts:

(1) The maximum bending moment is due almost entirely to the gas loads in one cylinder and occurs near top center on the firing stroke of that cylinder.

(2) Where there are a number of cylinders on the master rod, those cylinders the axes of which are at small angles with the master-rod-cylinder axis will contribute comparatively small moments.

Translating the foregoing recommendations into actual illustrations, the fact is evident that for seven-cylinder radials an investigation of cylinders Nos. 3, 4, 5 and 6 over a small angular range near their respective top-center positions on the firing stroke would be necessary. The master-rod cylinder is assumed to be No. 1 in every case. For a five-cylinder engine, an investigation of Nos. 3 and 4, and for a three-cylinder engine Nos. 2 and 3, would be sufficient. When the maximum turning-moment due to any cylinder alone has been found, the turning moments caused by any other cylinders that are on the compression or firing stroke at this particular point should be computed and added, algebraically. But any opposing moment to be added should not exceed that due to pumping, to allow for misfiring.

Legislative Regulation of Motor-Vehicles

(*Concluded from p.* 485)

I believe that we owe it to ourselves to study this paper carefully and compare it with previously printed data before we accept some of the low axle-loads suggested. Only a few years ago, when solid tires were general, loads of not less than 22,000 lb. per axle were advocated by this same agency. Was it a mistake in tests, computations or mere political expediency that has resulted in lower limits being reached? These lower limits seem to me hardly logical in the face of the improvement in tires, springs and general constructional features.

I agree, however, with the over-all dimensions as regards width and feel that for our committee to ignore the sound thoughts advanced on this point would be a calamity. Legislatures will not listen to an outside-width limit of more than 96 in., and we will nullify our own good thoughts if the manufacturers' representatives insist upon carrying this point.

While I admit the good work done by the Bureau of Public Roads and its able chief, let me point out the danger of a continual demand for Governmental assistance. I fear that in time the automotive industry will find itself in the same position as the railroads, which claim that they are in grave financial difficulties and therefore are asking a body of public men headed by Ex-President Coolidge to investigate the facts. Why is this necessary and what will be achieved by such an investigation when we already have a constituted body, the Interstate Commerce Commission, which represents absolute Government control over the railroads? This body has spent millions of dollars of public money on investigation, has established regulations governing the railroads, fixed the rates and says yes or no to executive decisions, yet we must contribute to the support of the railroads through the Reconstruction Finance Corp. The situation is a bewildering one that demands clear thinking.

Freedom of action in operation and trade relations has built up the enormous automotive industry. Restrictive laws and rulings by the Interstate Commerce Commission, despite huge founding subsidies, later wartime gifts and recent huge R. F. C. loans, find a great rail transportation industry supposedly on the verge of bankruptcy. In the face of such a situation, do we want more Government regulation? Or should we take steps to go back to former days by drawing the purse-strings tightly, thus choking off excessive public waste of funds that creeps into such operations?

THOMAS H. MACDONALD:—The need of a provision, of the sort suggested by Mr. Smith, to permit the occasional movement of poles, girders or other long and indivisible objects will be generally recognized and doubtless should be incorporated in the laws that are enacted. My impression is that some provision to the end mentioned is already made in most, if not all, States.

The recommendations made in the paper deal only with certain of the more important elements. No effort has been made to formulate a complete code.

To the comment by Mr. Winchester I can only rejoin that I do not recall that the Bureau of Public Roads has at any time advocated axle loading as great as 22,000 lb. for rural highways.